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DOE/NASA CONTRACTOR REPORT

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SOLAR HEATING AND COOLING SYSTEMS DESIGN AND DEVELOPMENT (Quarterly Report)

Prepared by

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Under Contract NAS8-32093 with

National Aeronautics and Space Administration
George C. Marshall Space Flight Center, Alabama

For the U. S. Department of Energy

U.S. Department of Energy

Solar Energy

FOREWORD

Honeywell was awarded Contract NAS8-32093 by the George C. Marshall Space Flight Center effective 9 July 1976.

The program plan calls for development and delivery of twelve prototype solar heating/cooling systems for installation and operational test at sites to be supplied by MSFC. Specifically, six heating and six heating and cooling units will be delivered, two each for single-family residences, multiple-family residences, and commercial buildings.

Lennox Industries, Marshalltown, Iowa, and Barber Nichols Engineering Company, Arvada, Colorado, are supporting Honeywell in subcontractor roles.

This document describes the progress of the program during the second three months (6 October 1976 to 9 January 1977). It is submitted to MSFC for information per DR500 Item 10.

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ILLUSTRATIONS

TABLES

SECTION I
SUMMARY

The Solar Heating and Cooling Development Program has cumulative negative schedule variance and no cost variance at the end of the first quarter, fiscal year 1977.

The negative schedule variance is a result of (1) an unplanned delay in initiating the Heating and Cooling Systems Design and Development Tasks, and (2) the delay in receipt of site specific data for the heating systems.

SECTION II
COSTS

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SECTION III
SCHEDULES

Figures 3-1 and 3-2 show progress in the Heating and Heating/Cooling portions of the program.

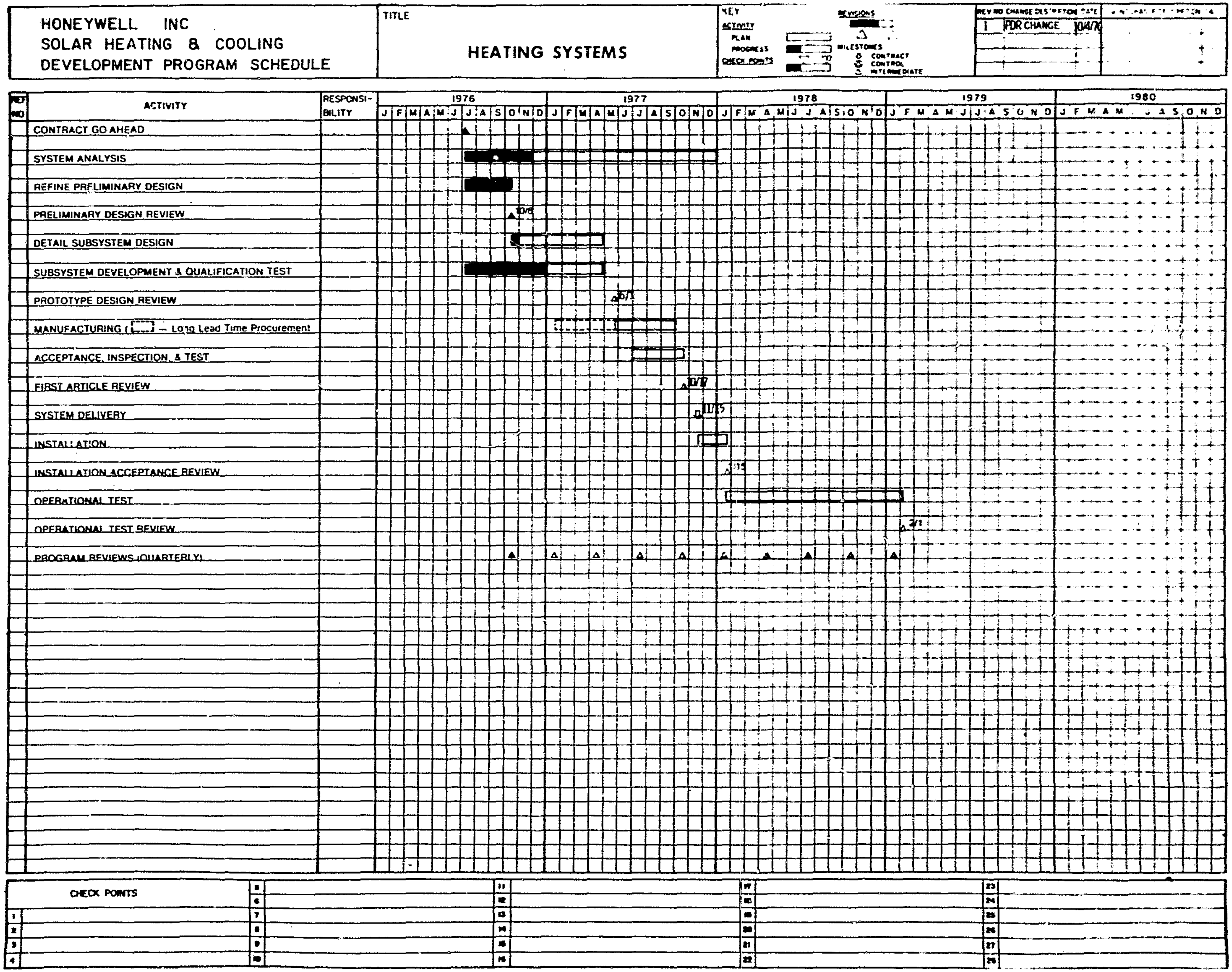

Figure 3-1. Heating Systems Development Program

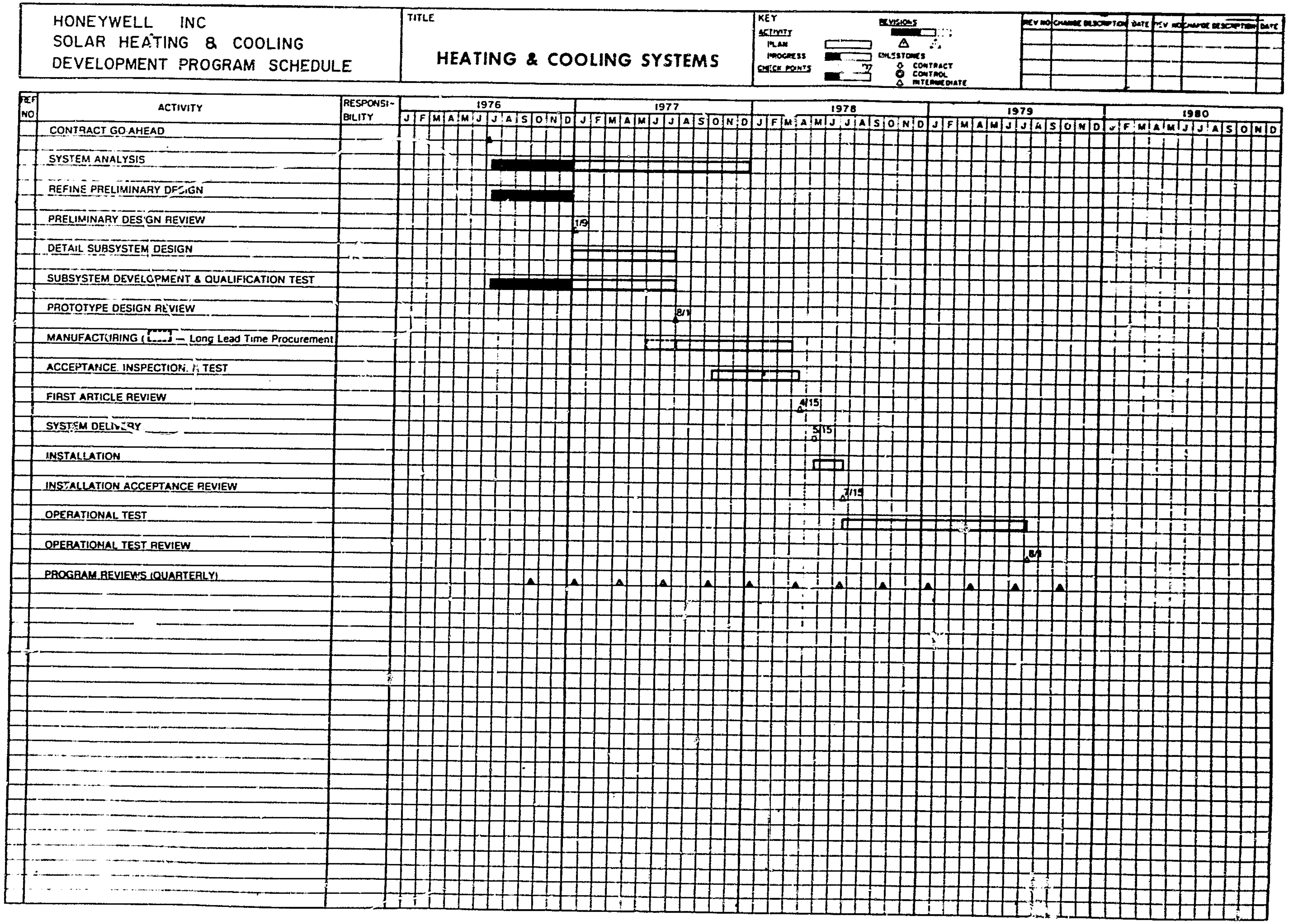

Figure 3-2. Heating and Cooling Systems Development Schedule

SECTION 4
TECHNICAL STATUS

4.1 TECHNICAL STATUS SUMMARY

The technical activity on this program since the first quarterly report has been primarily in the predesign of the Heating and Cooling Systems. This activity has been completed and presented at the Preliminary Design Review on January 18, 1977. The technical activity on the Heating Systems has been limited to further refinement/improvement of the multifamily and commercial systems configuration and generalized studies on several of the subsystems. These studies were in subsystems not only common to Heating/Heating and Cooling systems but in areas not affected by site requirements. The results of the above-mentioned activities are delineated in the following paragraphs.

4.2 SUBSYSTEM DESIGN AND DEVELOPMENT

The status and recent activity on the major subsystems of the Heating/Heating and Cooling Systems is delineated in the following paragraphs.

4.3 COLLECTORS

The collector subsystem is a product which is presently being manufactured by Lennox Industries and is considered developed to the prototype level. In addition, it was recognized that the collectors are modular, suitable for retrofit or new construction, and can be combined to provide any subsystem size. It was also recognized that a study of possible flow configurations for various collector arrays should be made. Of concern is the possibility of

nonuniform flow in the collector arrays which can lead to serious degradation of performance. A simulation program was developed from which guidelines will be established to determine the number, size, and arrangement of the collector modules and supply and return headers in the collector arrays. The results of this study will be used to design the collector arrays when specific site locations have been determined.

4.3.1 Parameters

A study of applicable Interim Performance Criteria (IPC) was made to determine what parameters of the solar collector to investigate. In the case of the collector, the development activity fell into two main areas: function and mechanical configuration.

During collector development, much effort was expended to design a highly efficient collector. Selective coatings and glass transmittance were investigated thoroughly. It was determined that the efficiency of collection was a fixed design characteristic at the beginning of qualification and that efficiency of a newly produced collector was not a relevant qualification criterion. At the same time, however, it was apparent that the effect of environmental exposure on efficiency had not been sufficiently investigated, so that efficiency degradation was determined to be a qualification criterion.

The mechanical configuration of the collector was investigated during development and yielded a design that permitted easy integration with the structure and a long service life. One element of the mechanical configuration which was not verified during development was exposure to mechanical loads inherent in a structural application.

As a result of reviewing the work done during development, it was determined that the following tests needed to be conducted to complete qualification of the collector:

- Degradation due to:
  - Solar exposure
  - Pollutants
  - Thermal exposure
  - Outgassing
- Mechanical loads due to:
  - Internal pressure
  - Roof loads
  - Hail

After determining the parameters to be tested, an investigation was made to find or develop suitable test techniques. Wherever feasible, the approach was to make use of existing standard test techniques. Toward this end, the IPC was used as a baseline to standardize the tests. Referenced test standards were used whenever practical. This approach resulted in the following series of tests.

4.3.2 Collector Subsystem Qualification Status

The collector subsystem will be qualified in part by a formal Qualification test in accordance with Document F3437-T-101. This test was initiated on 18 October 1976. It is progressing on schedule and completion is expected on or about 30 May 1977. Qualification status is currently as follows:

- Test Number 3.1, Pressure:

  Objective: Ascertain that the system does not leak under a hydrostatic pressure equal to 1.5 times normal collector working pressure.

  Status: This test is scheduled for Feb '77.

Plan: The setup to perform the test on the collector is ready as shown in Figure 4-1. The setup essentially consists of a pump and the collector. The pump will fill the collector with water at a pressure of 150 psig and maintain it at test pressure level for 15 minutes.

- Test Number 3.2, Service Loads:

Objective: Determine the ability of the collector to withstand a distributed load of 50 psf. This is to simulate typical roof loads of snow and wind.

Status: Complete.

Results: The collector withstands a uniform load of 50 psf with no damage or plastic deformation. A load of 78 psf applied for engineering information also resulted in no damage. The load consisted of uniformly distributed sandbags.

- Test Number 3.3, Hail:

Objective: To determine the ability of the collector to withstand impact of 1.25-inch hail at 82 fps without fluid leakage in the system.

Status: Complete

Results: No damage resulted to any component of the collector due to impact of 1.25-inch hailstone at 82 fps. This test level was increased for engineering information with the following results:

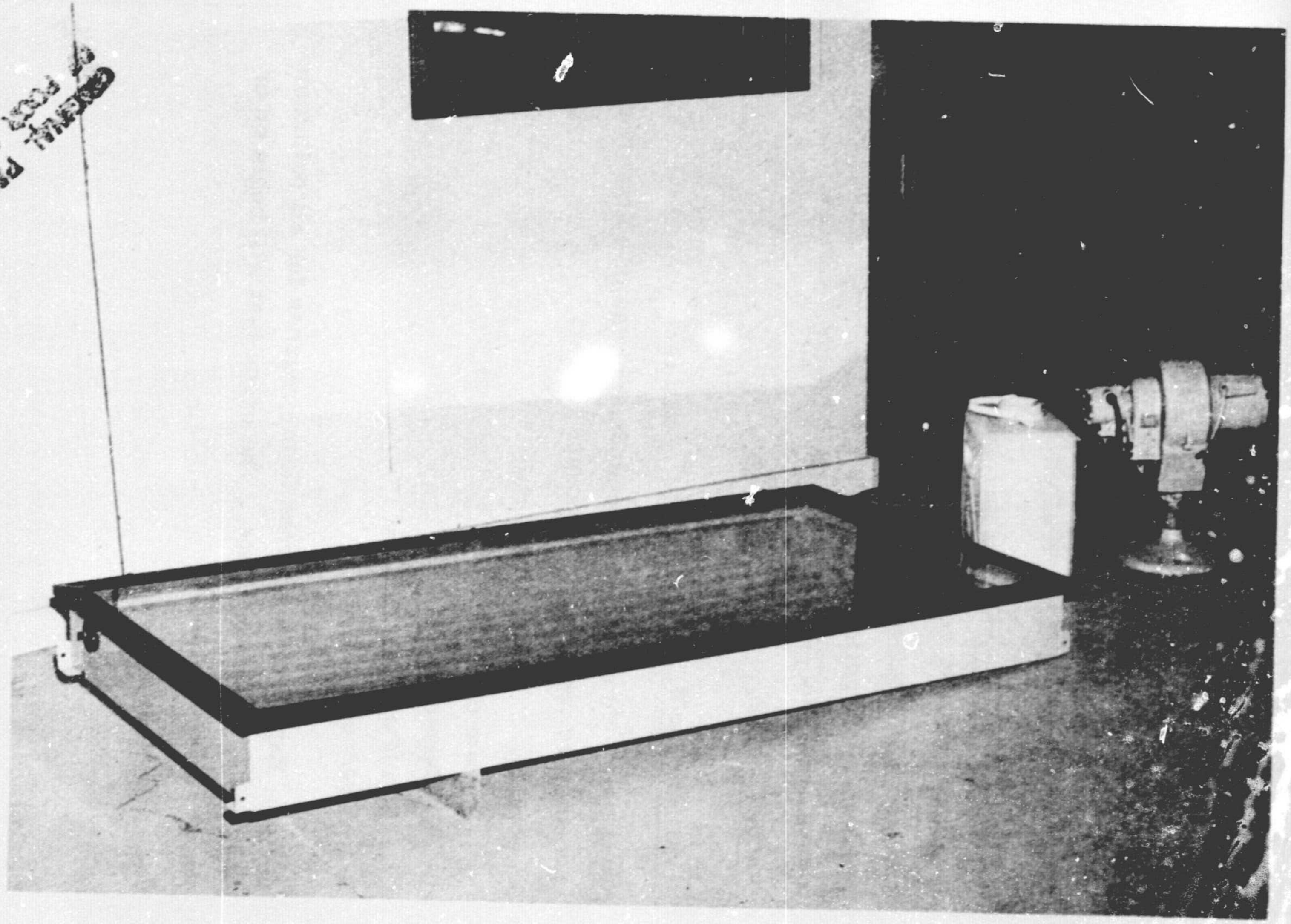

Figure 4-1. Experimental Setup to Perform Qualification Test 1.P.C.2.3.1 On the Solar Collector

2-inch hail, 107 fps - no damage
2.5-inch hail, 116 fps - glass breakage (outside cover)
3.0-inch hail, 131 fps - glass breakage

This test was conducted by firing an ice sphere from a compressed-gas launcher.

- Test Number 3.4, Solar Degradation:

Objective: To determine if degradation results from up to 6-month exposure near Phoenix, Arizona.

Status: Pre-exposure efficiency tests conducted on two collectors. Both collectors are now undergoing exposure at Desert Sunshine Exposure Tests, Phoenix, Arizona.

Results: None to date.

- Test Number 3.5, Pollutants:

Objective: To determine if coupon samples of collector components are degraded by exposure to:

Ozone
Sodium chloride
Sulferous acid
Nitric acid
Hydrochloric acid

Status: Exposure of coupons is in progress for all pollutants with the exception of ozone. The ozone test will begin on or about 15 January 1977.

Results: None to date.

- Test Number 3.6, Thermal Degradation:

  Objective: To determine if coupon samples of the collector are degraded by exposure to maximum service temperature.

  Status: Exposure of coupons is in progress.

  Results: None to date.

- Test Number 3.7, Insulation Outgassing:

  Objective: To determine if outgassing of organic materials results from day-night cycling of the collector.

  Status: Three cycles on the solar simulator are complete. One-hundred day-night cycles in Phoenix, Arizona, will begin on or about 15 January 1977.

  Results: Requirements met for three solar simulation cycles.

4.4 ENERGY STORAGE SUBSYSTEM

The collected solar energy which cannot be used for space heating or domestic hot-water heating at the time of its availability will be stored in the solar storage water tank. The stored energy would be available for future use (e.g., at night or on cloudy days). A system tank size of 1000 gallons represents reasonable size for residential use, while 8500 gallons seems feasible for multiple-family and commercial use. The flat heat tank is sized to store excess solar energy throughout the year without water boiling in the tank.

The actual water temperature in the storage tank will vary depending upon the energy requirements for space heating/cooling and domestic hot-water heating. The storage system will use stratification as much as possible for

maximum utilization of the collectors and stored energy. Valving will permit storage charging and recharging with the activation of the storage pump and proper control signals. A layer of insulation will reduce heat losses. Actual location and size of the storage tank will be determined after specific sites have been selected.

4.5 SPACE HEATING SUBSYSTEM

Space heating for residential and multiple-family use is provided using a forced-air recirculating system. The fan is integral to the commercially available auxiliary energy furnace. Heating of the air from the solar energy source occurs through heat exchange in a hot-water coil which is installed in the return-air cabinet. Coil capacities of 45,000 to 100,000 Btu/hr are available.

The commercial units for space heating/cooling are provided using a forced-air recirculating system with fresh-air makeup. The fan(s) is integral to the rooftop-mounted auxiliary energy furnace. Heating of the air from the solar energy source occurs through heat exchange with a hot-water/chilled-water coil which is installed in each rooftop unit. The cooling capacity of each coil is 300,000 Btu/hr.

4.6 AUXILIARY ENERGY SUBSYSTEM

The residential and multiple-family gas-fired furnace (Figure 4-2) is equally applicable to residential and small business or commercial installations. The low height, quietness of operation, and modern cabinet design permit

installation in a recreation or family room, basement, utility room, or closet. A low-boy-type installation is made possible with the addition of a return-air cabinet to the high-boy furnace. The low height of the furnace, due to the design features of the heat exchanger, will allow ample space for basement installations. The return-air cabinet can be installed on either side of the furnace. Direct expansion evaporator units and matching condenser units, electronic air cleaners, and power humidifiers can be added for a complete, all-season total comfort installation. Quiet operating blowers have sufficient capacity to handle all air volume requirements.

The commercial unit (Figure 4-3) will be a modification of an existing commercial rooftop unit of 500,000 Btu/hr heating capacity. The number of rooftop units required will be determine after site specific information is available.

The combination of gas-fired heating and cooling units with bottom handling of conditioned air are designed primarily for rooftop installation with optional fresh-air intake. A separate roof frame mates to the bottom of the unit and when flashed into the roof permits weatherproof duct connection and entry into the conditioned area. No additional roof curbing or flashing is required. The single-package unit can also be installed on a slab at grade level with end-handling of conditioned air. The insulated single cabinet houses gas-fired heaters, belt drive blowers, air filters, and the optional air intake dampers which are shipped complete with all controls wired.

The development status of the auxiliary energy and space heating subsystems is as follows:

- <u>Residential-Multiple-family Units</u> -- Since the space heating subsystems will be the same for single-family and multiple-family applications, and the auxiliary energy subsystem will also be the same other than the size, they will be treated the same as residential units.

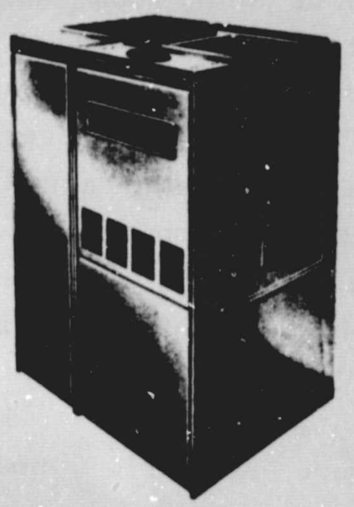

Figure 4-2. Lennox Gas-Fired Furnace

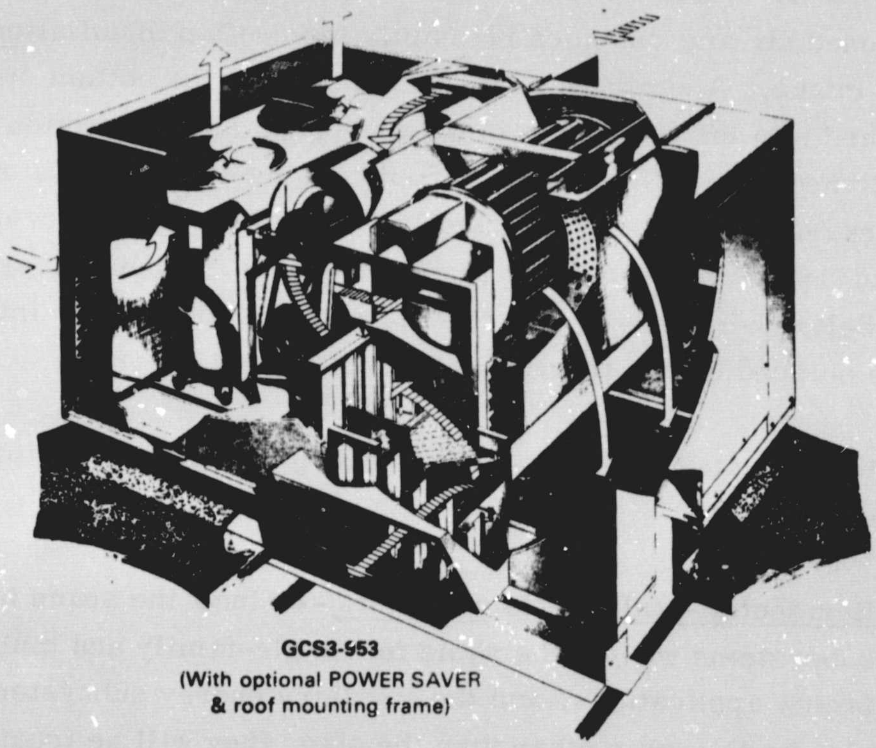

Figure 4-3. Lennox Rooftop Heating/Cooling Unit

Lab tests have concluded that based on Seely tests (IPC No. 2.1), the existing furnace motor will withstand a return-air temperature of 140°F. It appears that a special high-temperature motor will not be necessary. The operational characteristics of the furnace have also been test-verified as unaffected when integrated with the space-heating subsystem. These tests are described in the following paragraphs.

A test setup was made as shown in Figures 4-4 and 4-5. The space heating subsystem coils were simulated with electrical heating coils shown in Figure 4-6 and a perforated plate downstream of the heating coils. The combined flow resistance of the electrical heating coils and the perforated plate was the same as it would have been in the presence of the space heating subsystem coils. The heat dissipated from the electrical resistance coils was varied with a power rheostat to simulate the variations in an actual solar subsystem due to changes in water flow rate, and the temperature of the solar heated water. The return air temperature (T_R) at the inlet to the blower and the supply air temperature (T_S) at the exit of the outlet duct were measured with thermocouple grids. The static pressure rise (P_S) across the furnace was measured by connecting an inclined manometer to pressure taps as shown in Figure 4-7.

The voltage to the electrical resistance coils was increased in steps, thus giving the temperatures time to stabilize. The temperatures T_R and T_S were noted for a constant static pressure rise of 0.50 inch of water until fan limit control switch cut off the power supply to the heating coils at T_S = 200°F. Figure 4-8 shows the variation of the supply air temperature (T_S) with respect to return air temperature (T_R). Based on this test, it was concluded that in an actual solar subsystem, the

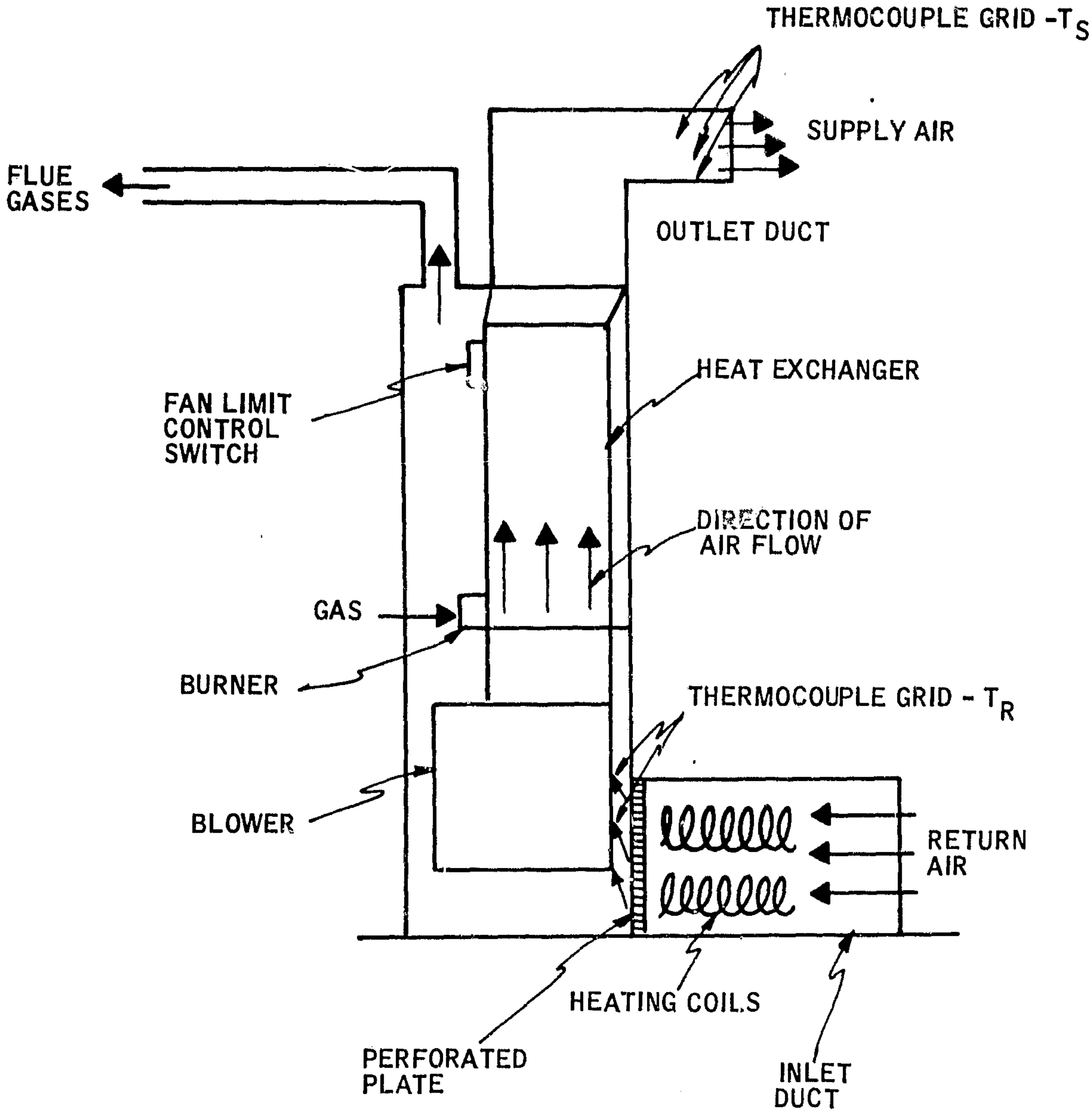

Figure 4-4. Simulated Test Setup to Study the Operational Characteristics of Residential Heating Units

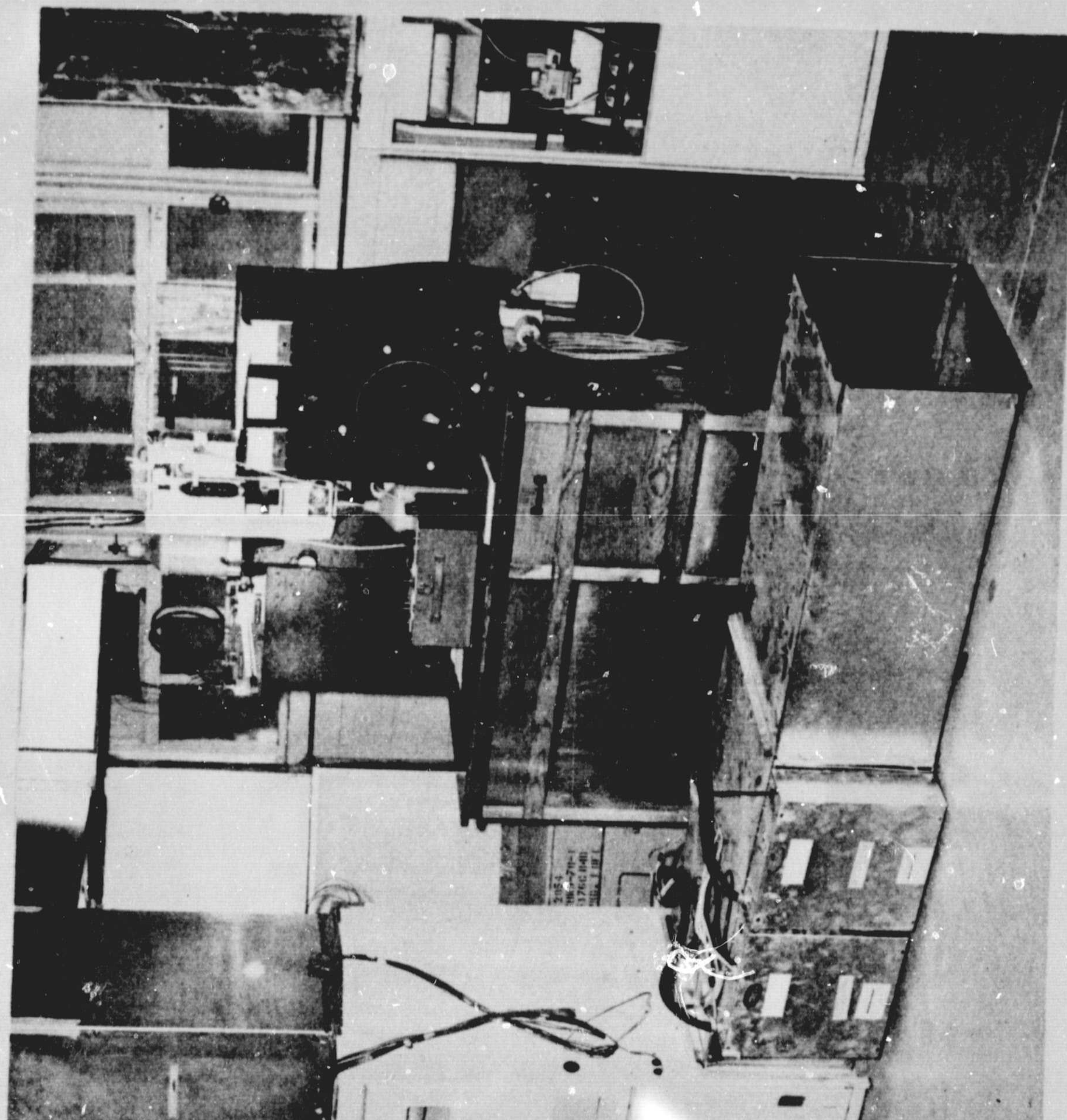

Figure 4-5. Residential Heating Unit Testing Setup

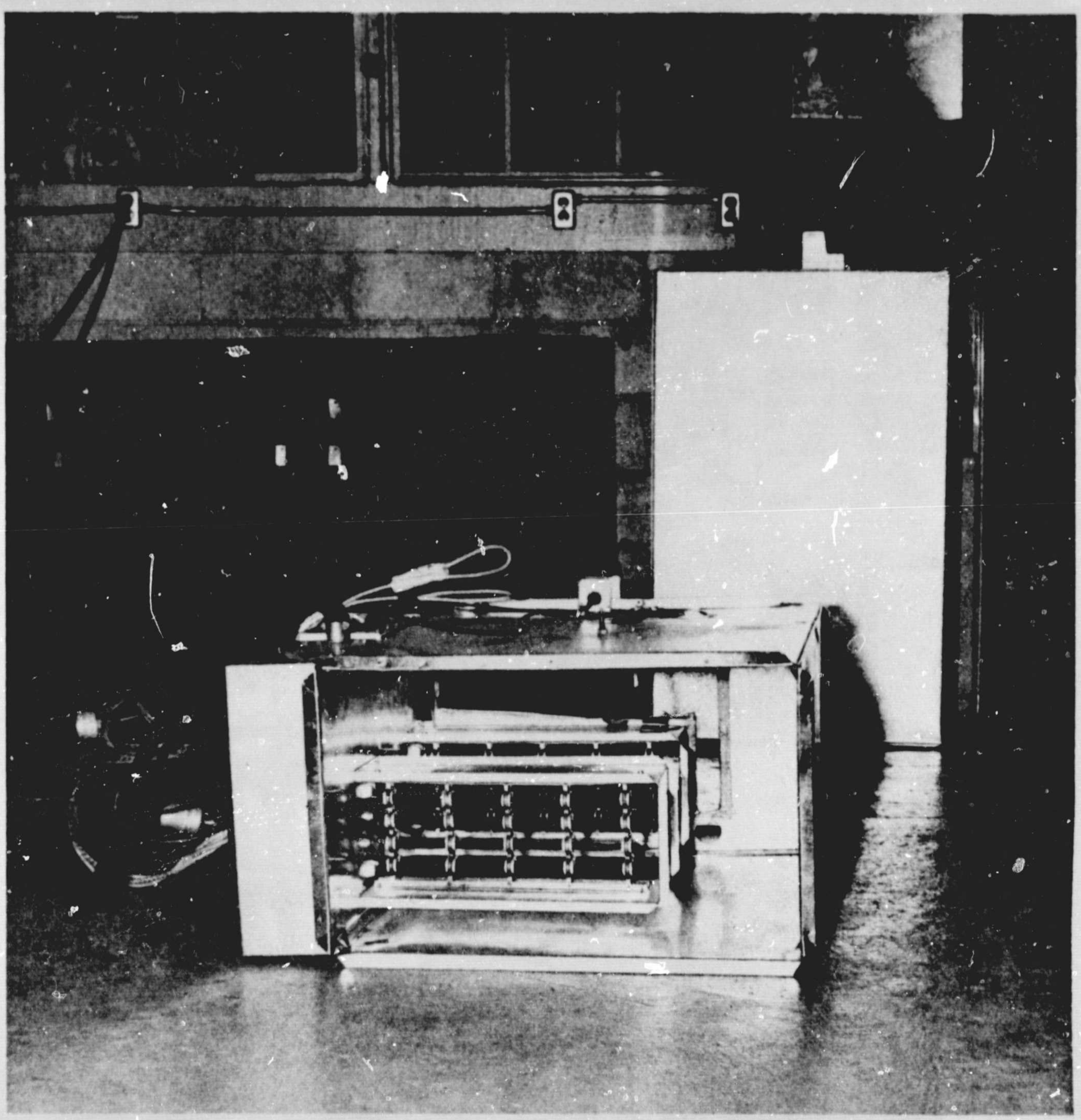

Figure 4-6. Electrical Resistance Heating Coils Used During Tests

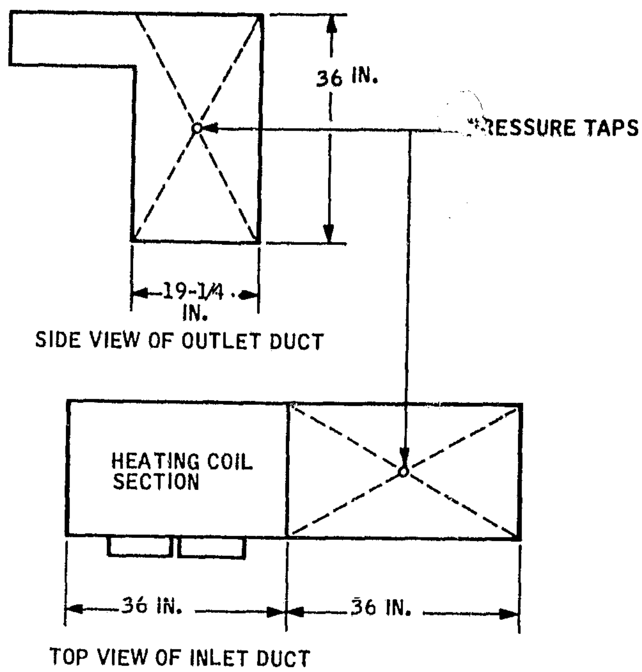

Figure 4-7. Location of the Pressure Tape

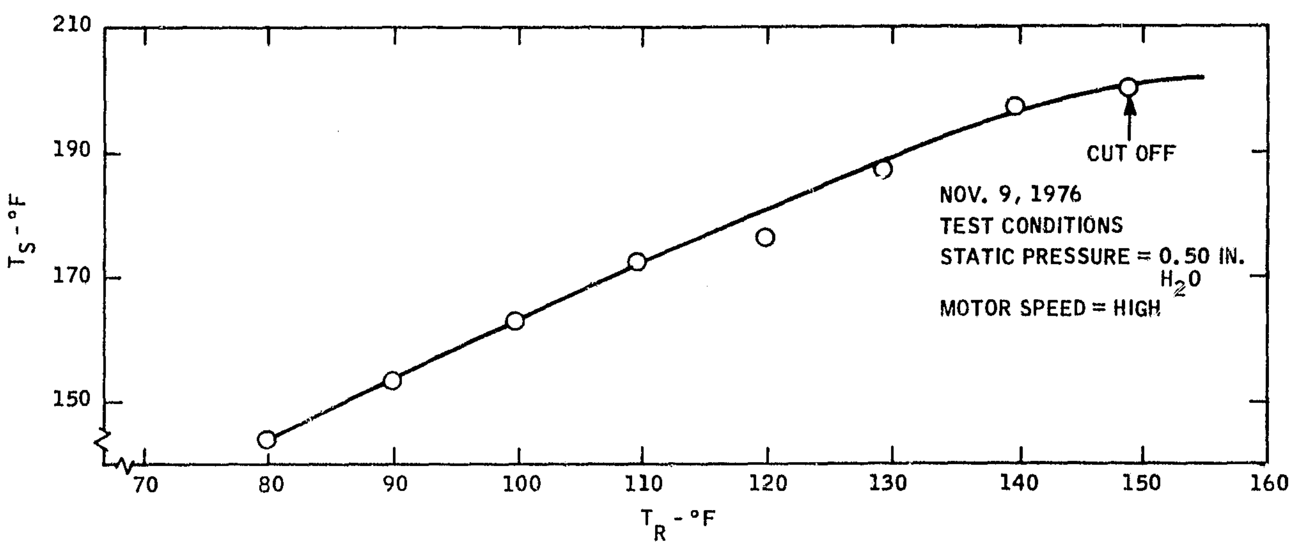

Figure 4-8. Variation Supply Air Temperature (TS) with Respect to Return Air Temperature (TR)

fan limit control switch will turn off the furace gas valve when $T_S = 200°F$ (@ $P_S = 0.50$ in. H_2O) in accordance with A.G.A. standards. It was concluded from these results that the return air temperature could go as high as 150°F.

The only deviation required on the furnace will be a wiring change which will turn on the furnace blower when the thermostat calls for first-stage solar heating. This eliminates the need for a sure-start fan control.

The detailed drawings of the cabinet for the space heating subsystem have been completed. Assembled and sectional views of the cabinet are shown in Figures 4-9 and 4-10. The prototype cabinet and space heating coil have been constructed for this subsystem. Figure 4-11 shows the photograph of the cabinet along with the space heating hot water coil. The assembly of the space heating subsystem will be ready and verification tests will be run in the near future.

- Commercial Units -- A rooftop unit will be used to adapt a water coil located upstream from the blower and gas-fired heat section. The same coil will be used for both hot-water heating and chilled-water cooling. Efforts are being made to size the coil and lay out the coil location in relationship to a GCS3-1853-500 unit. Plans also call for investigating the possibility of using a new line of Lennox rooftop DSS1 units for the commercial application if the time required for development of this new line is feasible.

Based on the market analysis, it is clearly indicated that there is no need for heating-only commercial applications; this would make it advantageous to use the DSS1 units in the commercial application because of the possible time savings involved.

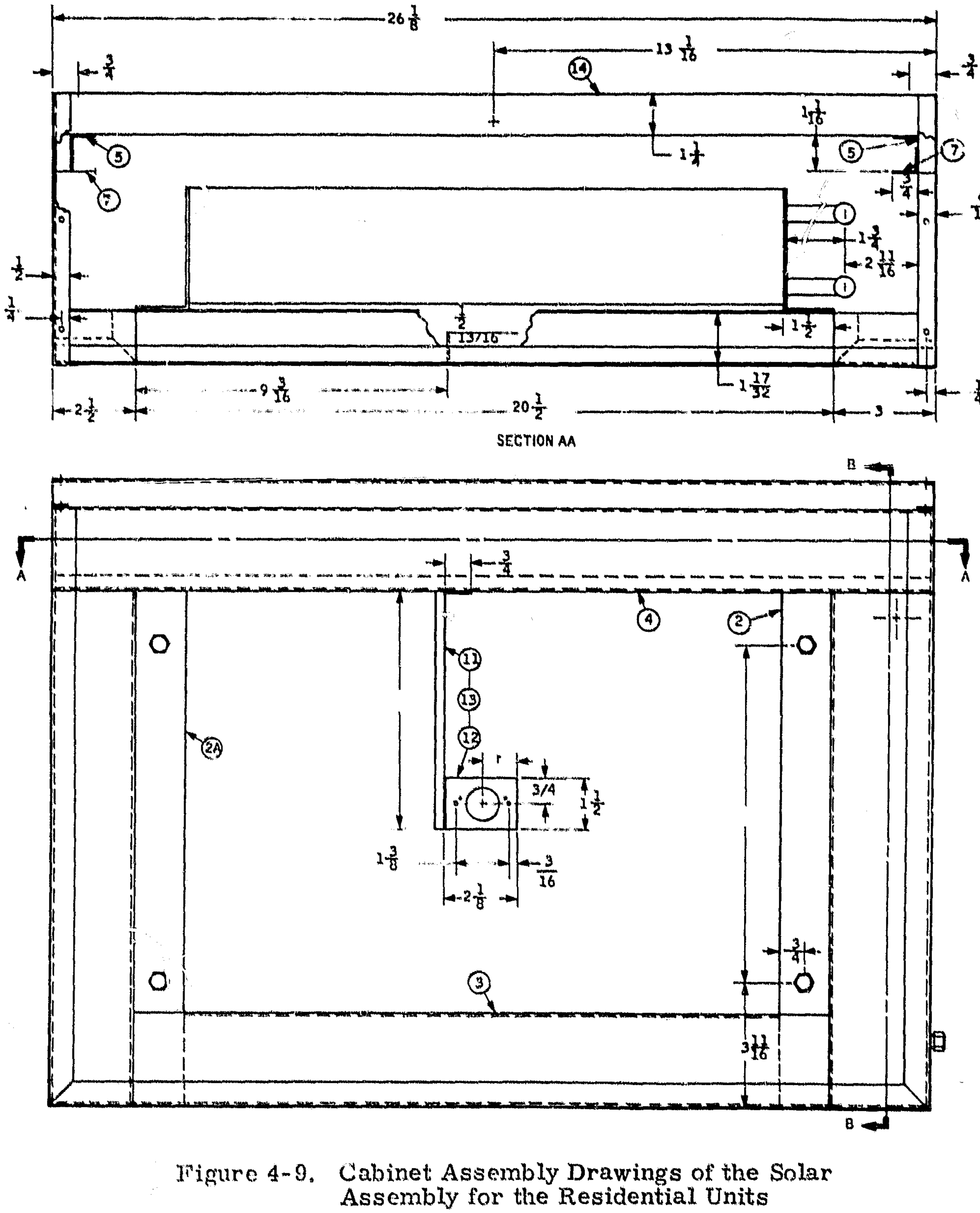

Figure 4-9. Cabinet Assembly Drawings of the Solar Assembly for the Residential Units

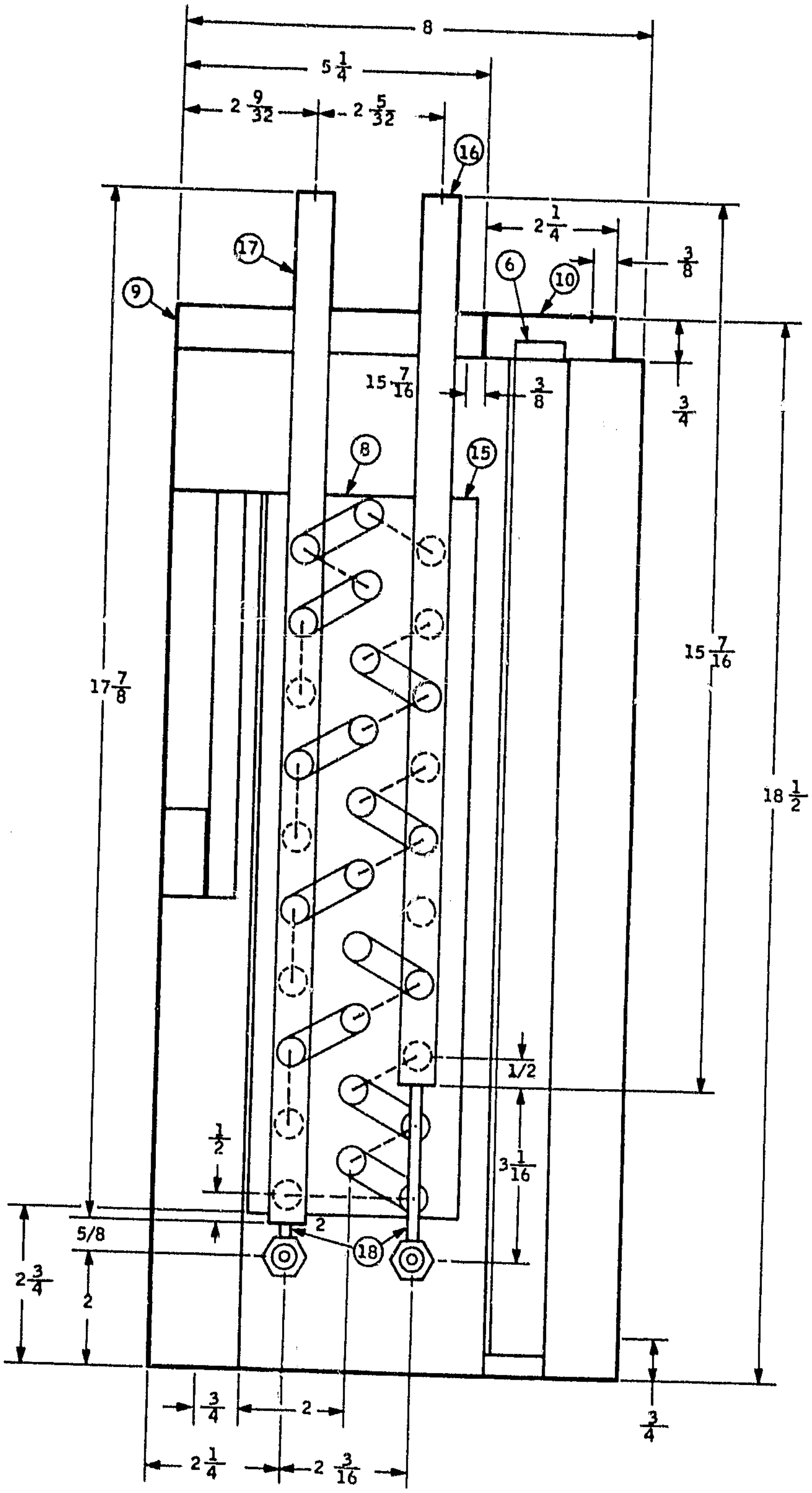

Figure 4-10. Section View at BB (Reference Figure 4-9)

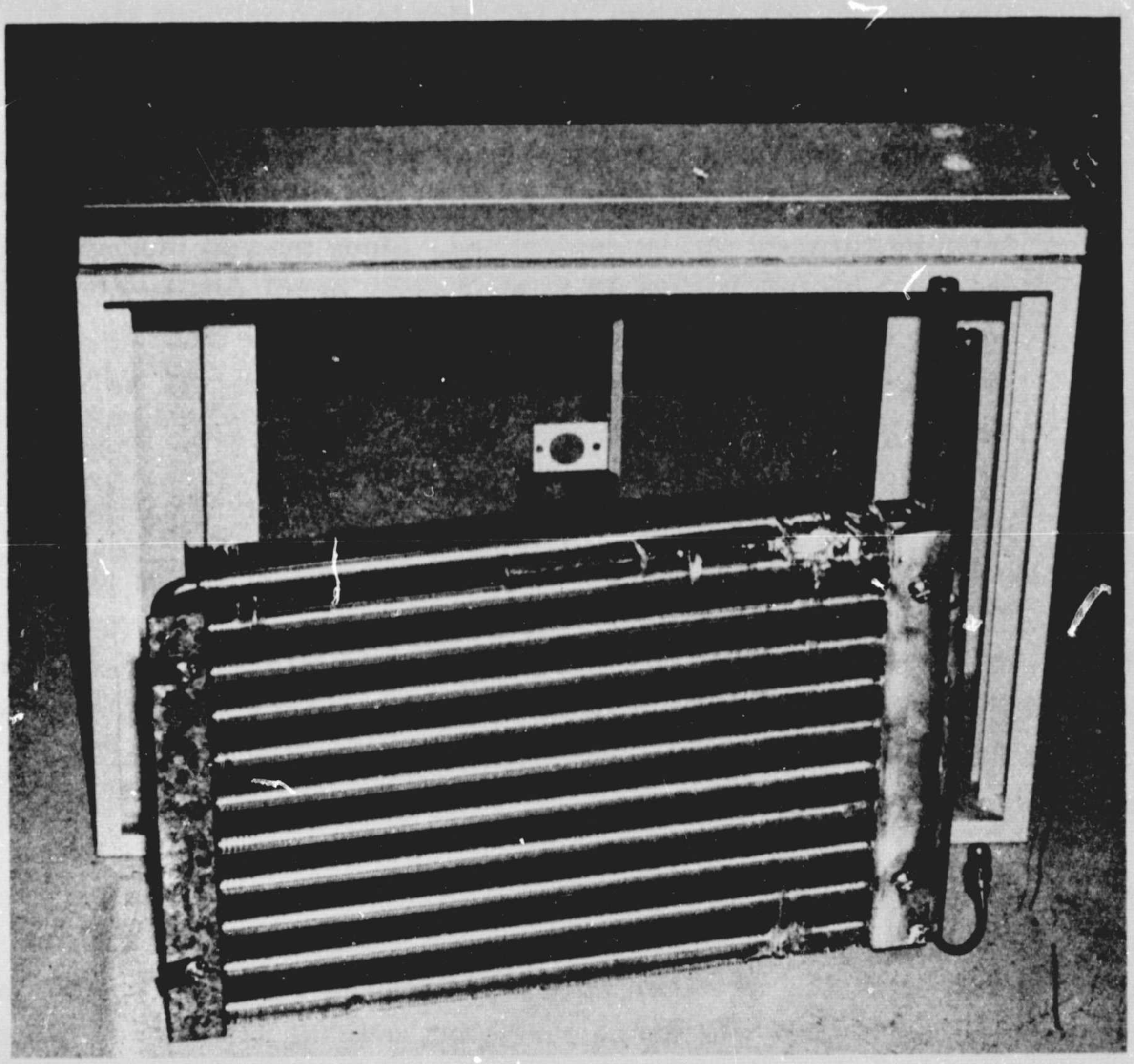

Figure 4-11. Space Heating and Coil of the Space Heating Subsystem for Residential Units

The coil will be sized for the cooling cycle and the same coil would likely turn out to be oversized for the heating cycle. However, in this situation, it is better to have only one size coil.

Again, the motors will be test-verified to ascertain that they will withstand 140°F air flow-through. Further, the air-handling capacity will be determined. Since the coil is upstream from the gas-fired heat exchanger in chilled-water cooling operation, condensation could occur in the heat exchanger. A provision for draining of condensate will be made.

4.7 HOT-WATER SUBSYSTEM

The hot-water subsystem consists of two hot-water heaters in series. The first heats the incoming domestic water with solar energy, while the second boosts the water to the final deliverable temperature, if added heat is necessary.

The solar energy which cannot be used for space heating will be used by the domestic hot-water preheater by using heat from the solar water storage tank or water preheating as shown in Figures 4-3 and 4-4. The water temperature in the preheater will not be controlled and thus will vary as domestic water use and the storage tank temperature vary. The inlet to the preheater will be city water. The outlet from this preheater will go to the input side of the domestic hot-water heater through a three-way control valve. The components of this subsystem are:

- Solar water-to-water preheater

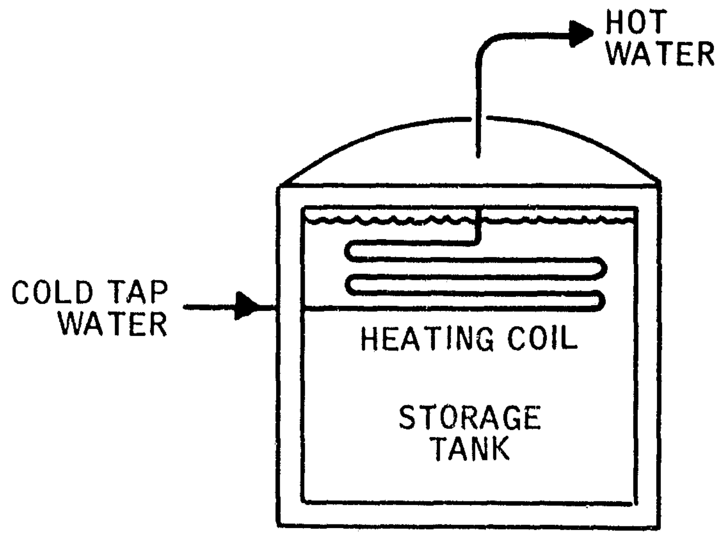

Figure 4-12. Solar Preheat DHW Subsystem for Single-Family Residence Applications

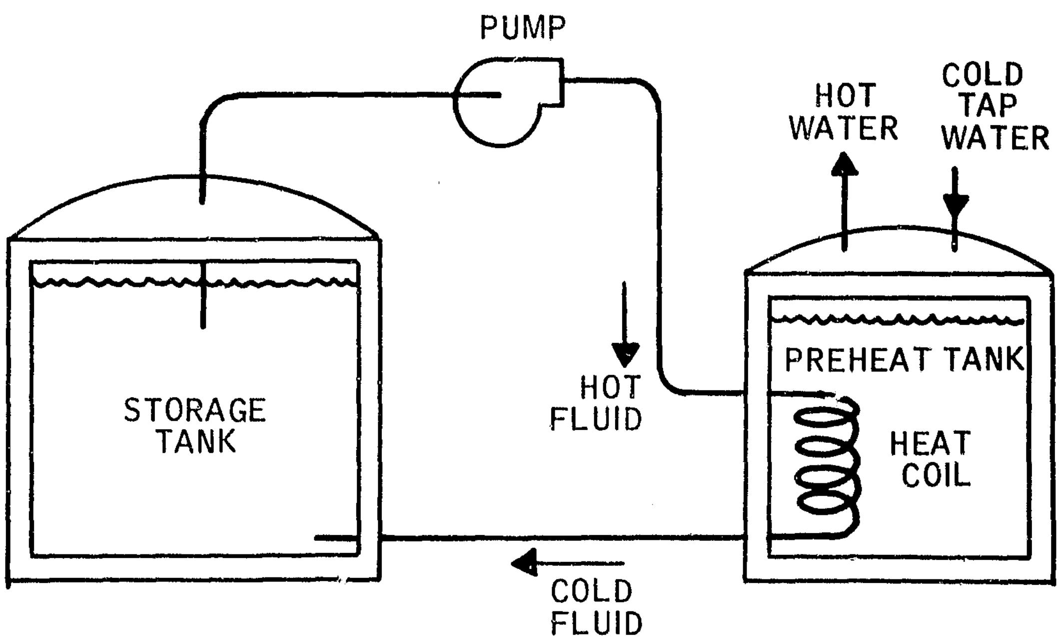

Figure 4-13. Solar Preheat DHW Subsystem for Multiple-Family Residence and Commercial Applications

- Gas-fired hot-water heater
- Self-contained three-way control valve

The domestic hot-water heater is sized using the requirements of the specific site.

The system can be modeled by using processes operating batchwise and involving heat transfer in determining the average temperature differences to describe the phenomena. The solar water preheat subsystem falls into this category (simple tank with heating coil). Figures 4-12 and 4-13 show schematics of the solar preheat DHW subsystems for single-family residence, multiple-family residence, and commercial applications.

Figure 4-14 shows the temperature rise of domestic water through a coil as a function of UA values for various tank temperatures. Figure 4-15 shows the same data plotted as coil outlet temperature as a function of coil length for various coil sizes for one tank temperature. Assuming a 50-foot coil of 0.75-inch copper tubing in a 140-gallon tank, a 55°F temperature rise would result. This corresponds to a UA of about 1100. Figure 4-16 shows pressure drop for various coil sizes. For the single-family unit, the pressure drop is insignificant. Figure 4-17 shows additional data on the 0.75-inch preheat coil. A storage tank temperature of 170°F would provide all of the 140°F domestic hot-water required for a flow of 2 gpm.

For the multiple-family and commercial applications, the system essentially "reverses its function." The coil is in a domestic water preheat tank, and the concern is how rapidly this tank can recover for various fluid temperatures, flow rates and coil UA values. Figure 4-18 shows the relationship between the recovery rate of the 326-gallon tank with heating water at 160°F as a function of the UA for various heating water flow rates. Figure 4-19 expresses the same data as Figure 4-18 for a 0.75-inch coil of various lengths,

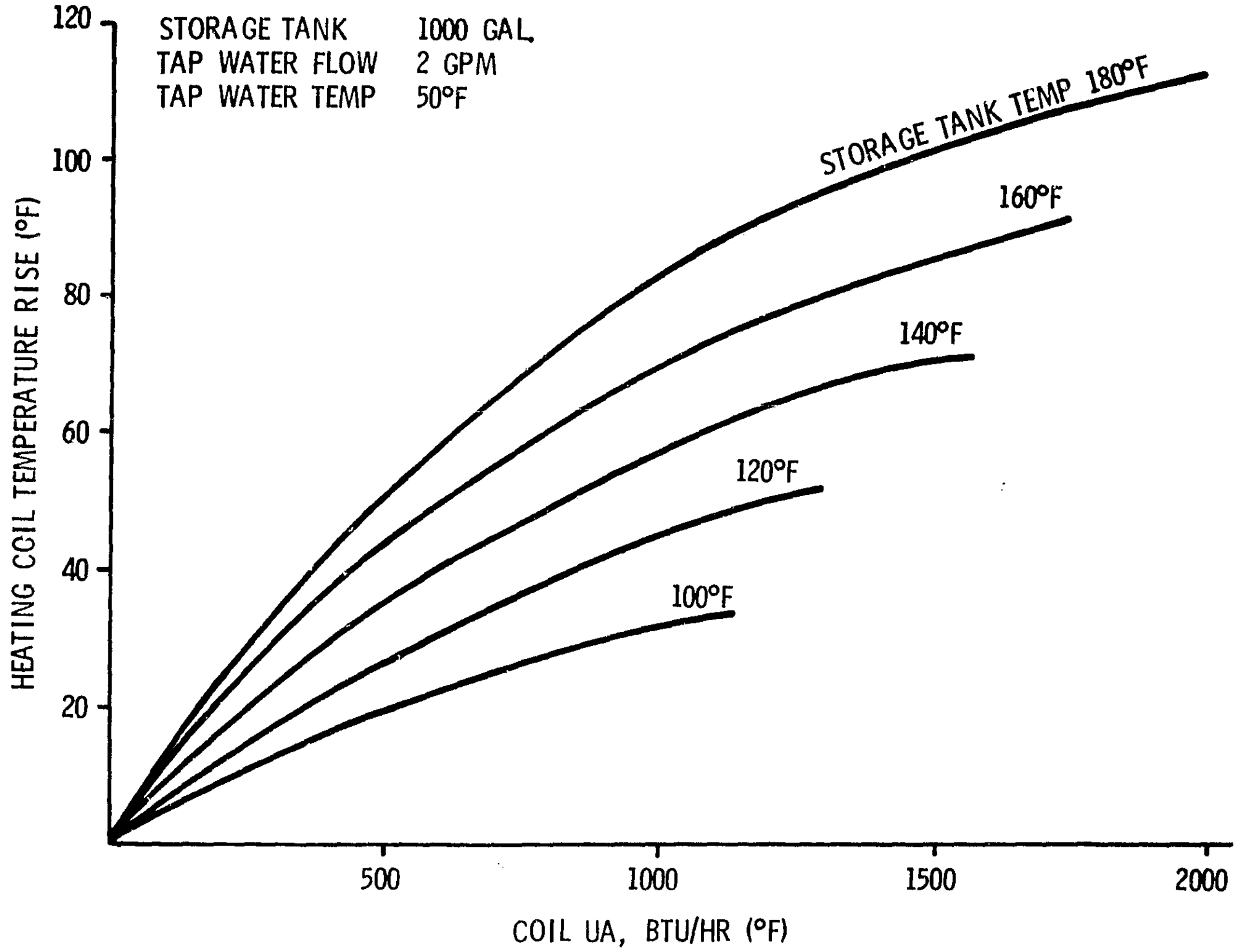

Figure 4-14. DHW Temperature Rise for Single-Family Residence Preheat Subsystem

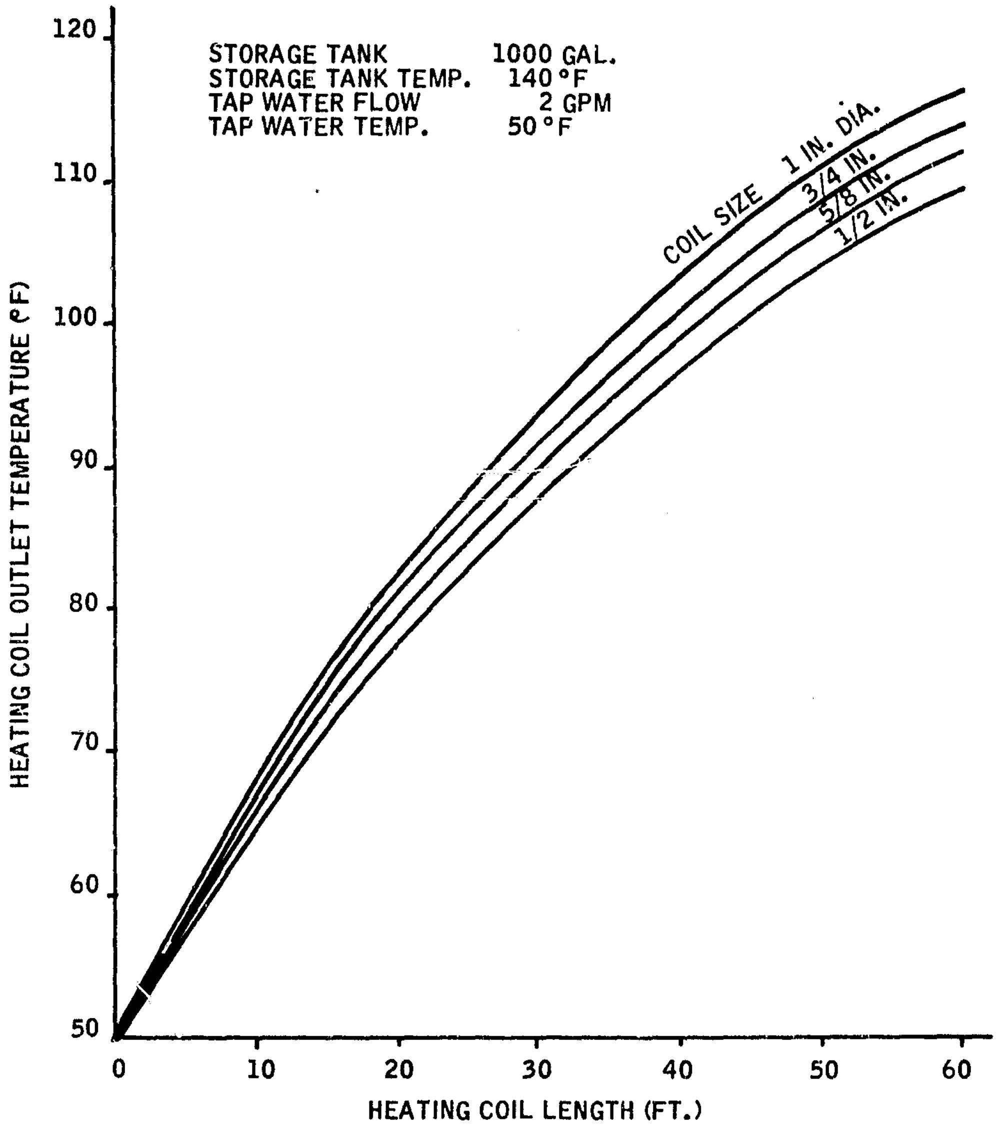

Figure 4-15. DHW Temperature Rise for Single-Family Residence Preheat Coil

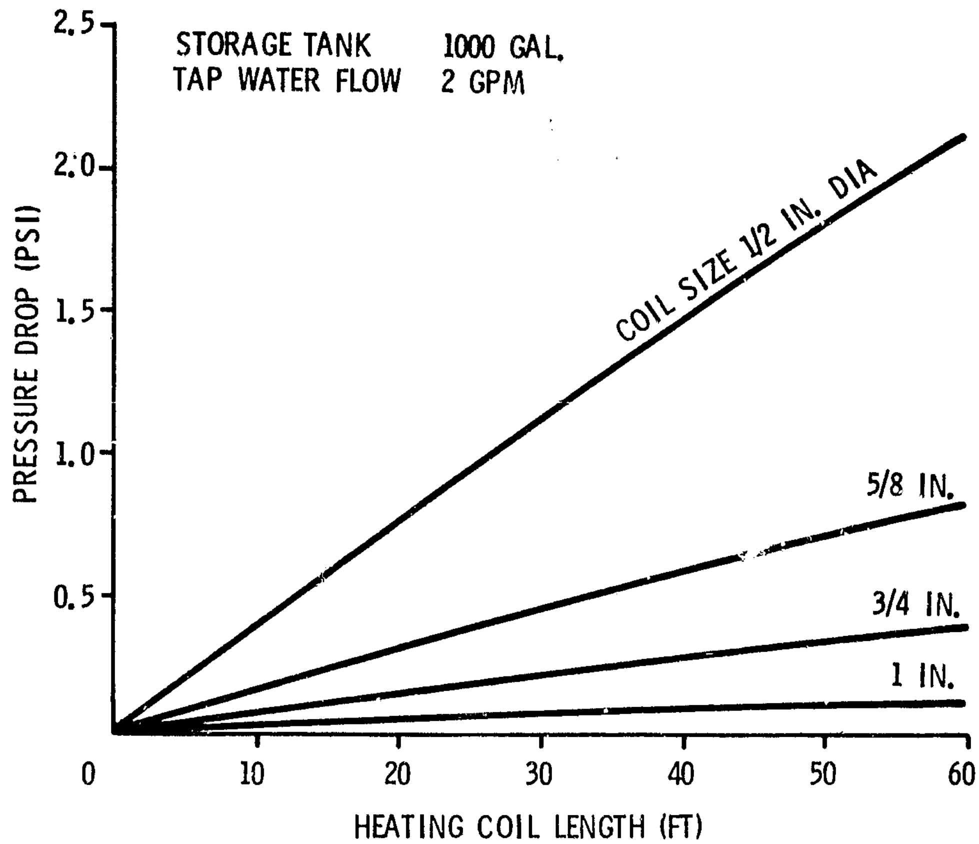

Figure 4-16. Pressure Drop for Single-Family Residence Preheat Coil

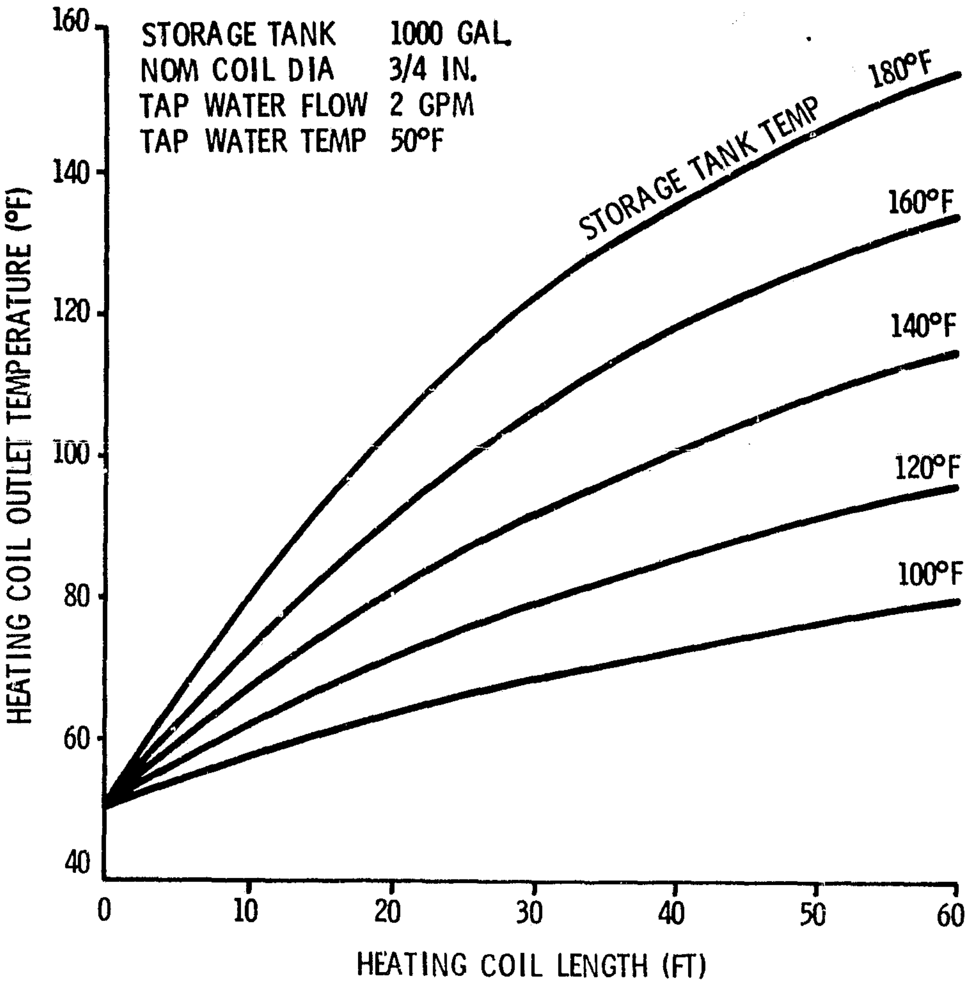

Figure 4-17. DHW Temperature Rise for Single-Family Residence 0.75-inch Solar Preheat Coil

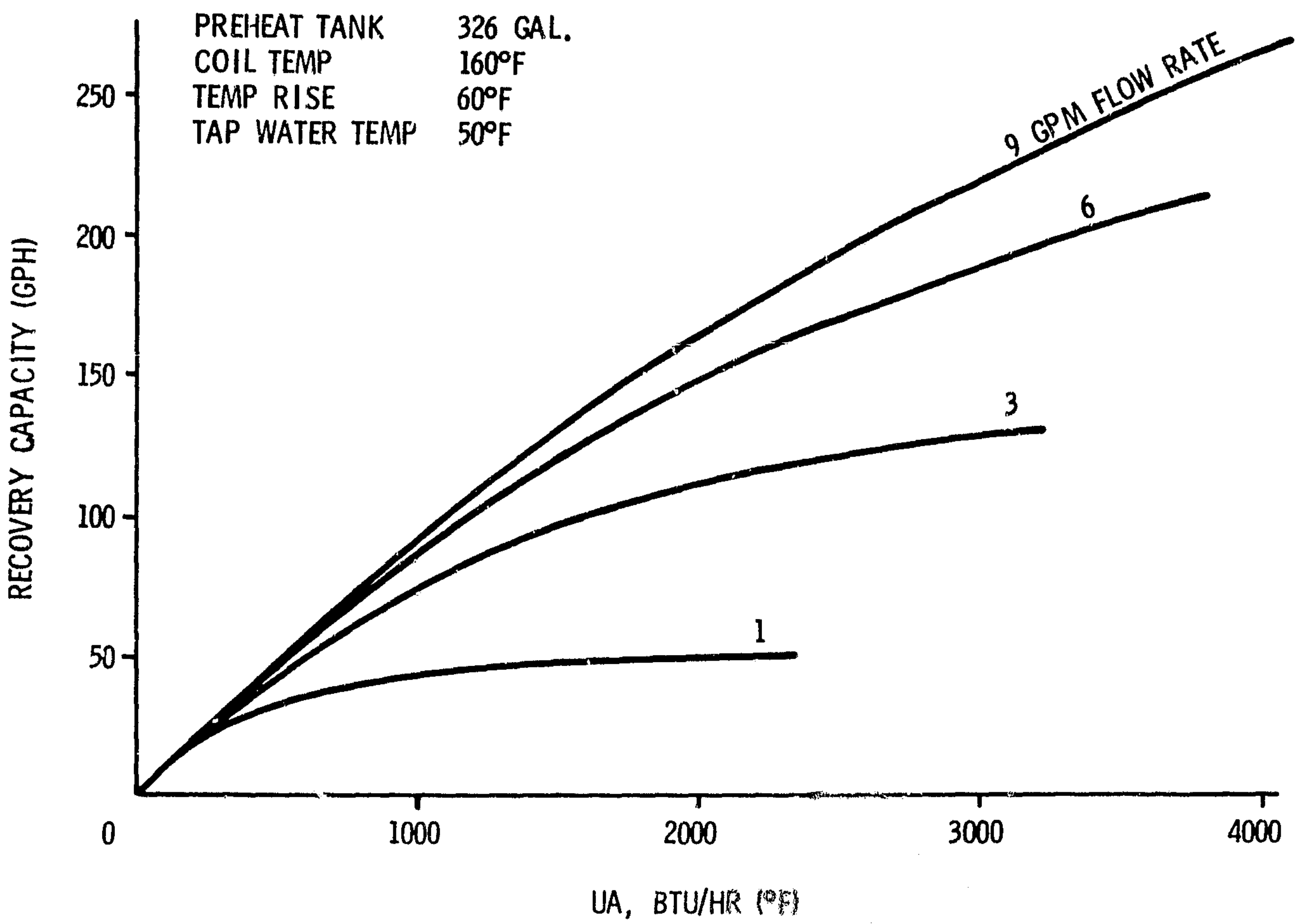

Figure 4-18. Recovery Capacity of DHW in Multiple-Family Residence Preheat Subsystem as Function of Heating Coil UA

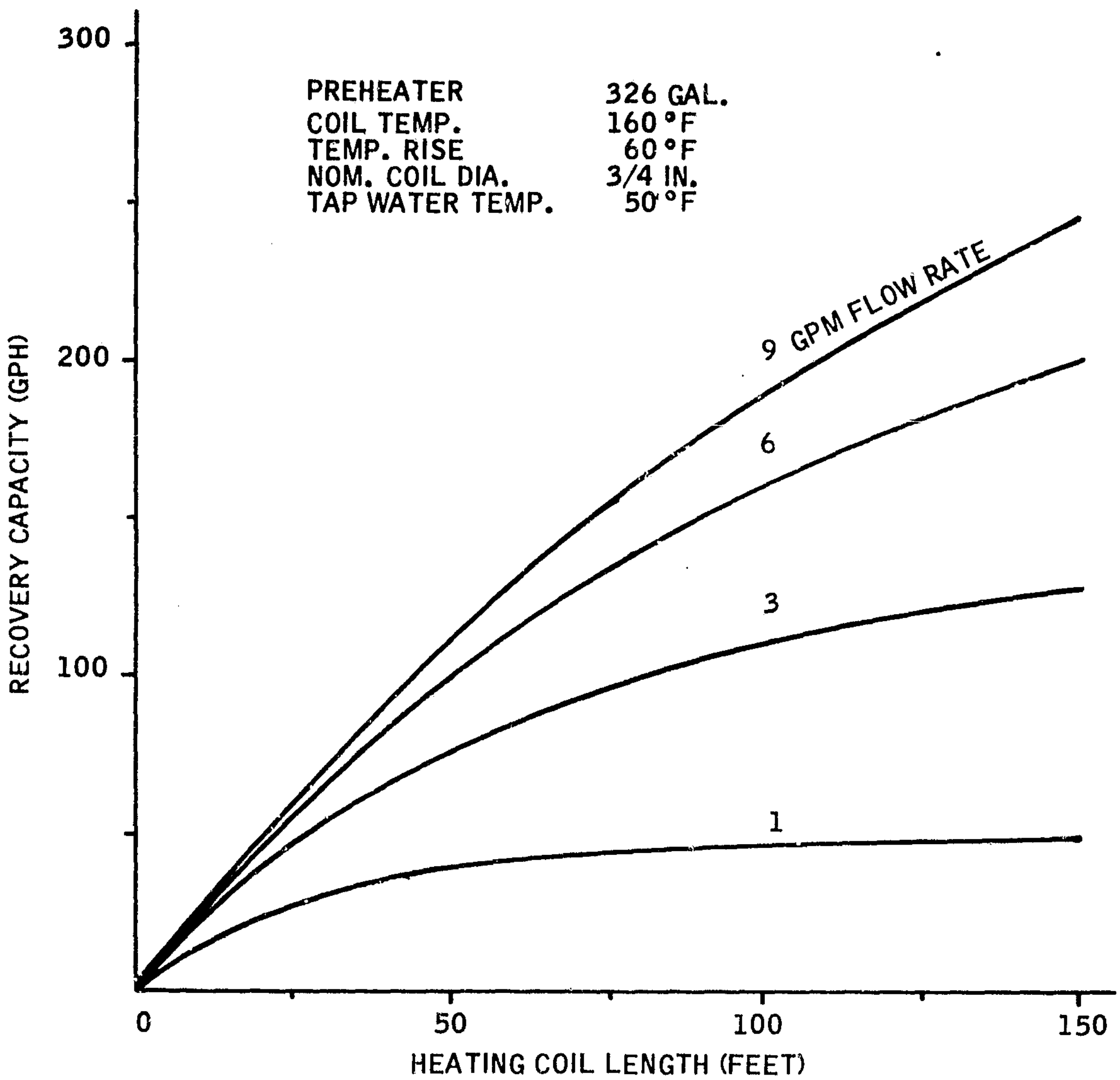

Figure 4-19. Recovery Capacity of Heating Coil in Multiple-Family Residence Application as Function of Coil Length

while Figure 4-20 gives the recovery capacity as a function of flow rate. Figure 4-21 can be used for predicting performance of a 0.75-inch heating coil of 150 feet effective length. Commercially available units with recovery rate of 300 gallons/hour are available.

The proposed system for single-family units will be as shown in Figure 4-12. During detailed design, tradeoffs between cost, performance, and size will be updated and finalized, while the commercial and multiple-family units will use commercially available units such as Patterson-Kelley or Bell and Gosset water-fired preheaters.

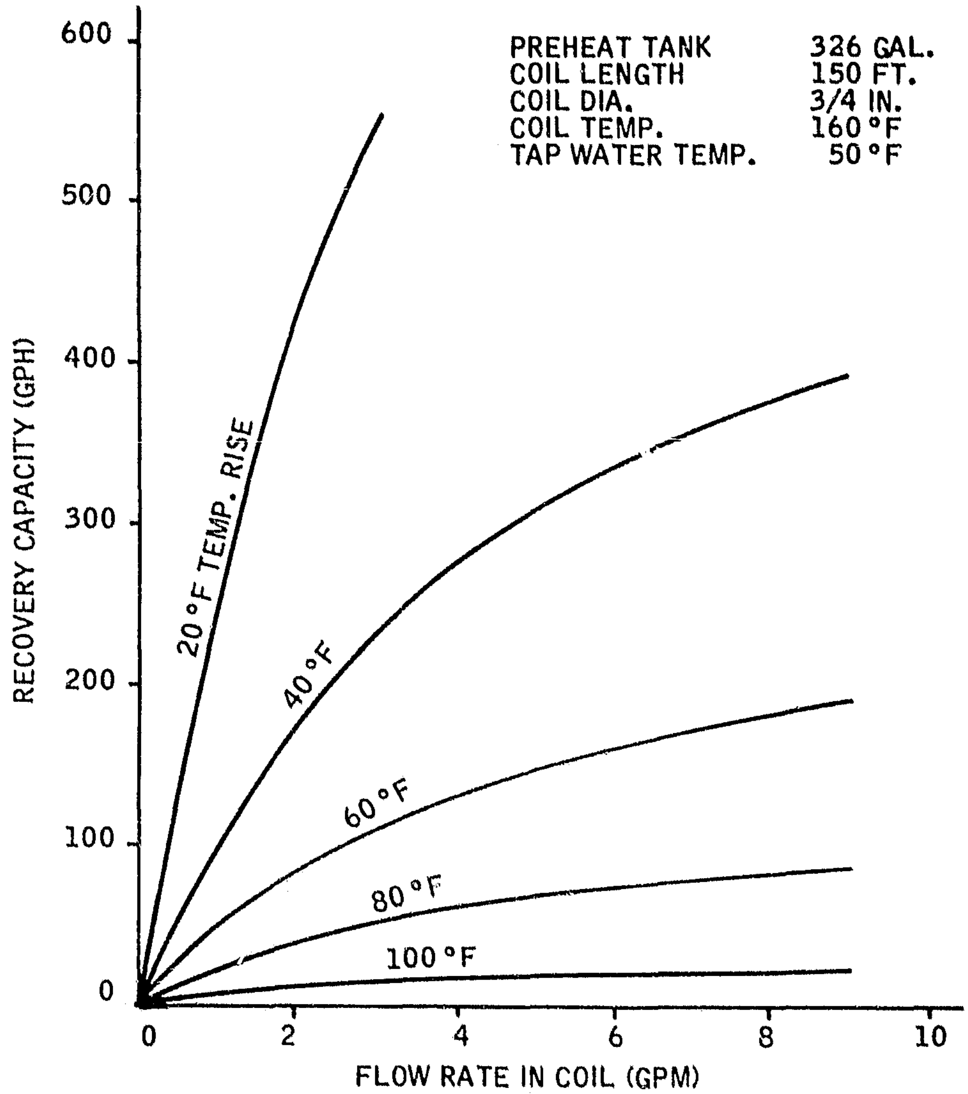

Figure 4-20. Recovery Capacity of Heating Coil in Multiple-Family Residence Application as Function of Coil Flow Rate

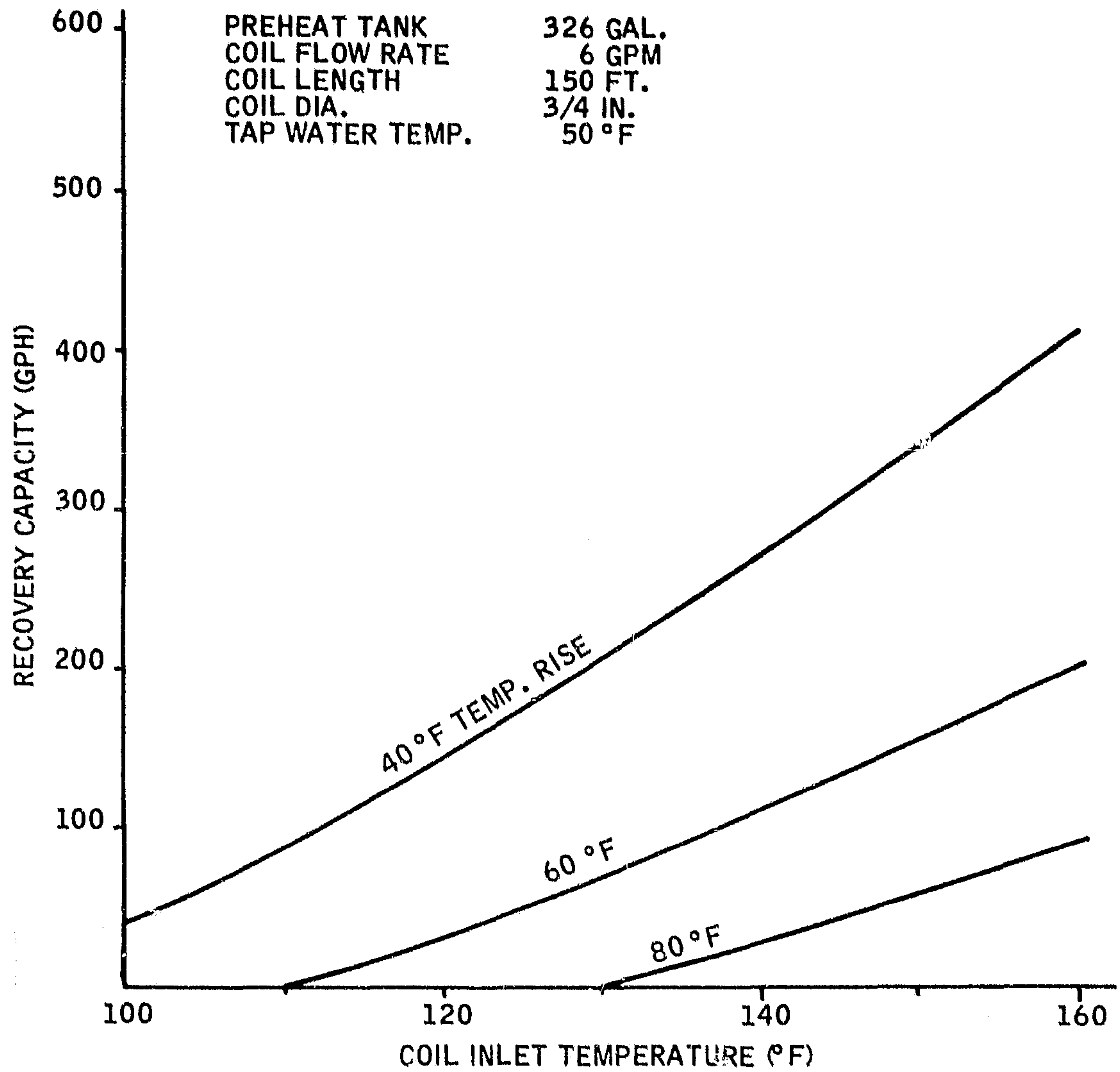

Figure 4-21. Recovery Capacity of Heating Coil in Multiple-Family Residence Application as Function of Coil Temperature

4.8 ENERGY TRANSPORT SUBSYSTEM

The energy subsystem consists of components necessary to ensure that subsystem components interface with one another and that necessary components are supplied for safe and efficient operation of the system. The following components are included in the transport subsystem:

- Piping sized for the system and site
- Expansion tank with fittings, ASME-rated
- Air separator and pressure-relief valves, ASME-rated
- Flow control devices with flowmeters
- Circulating pumps, centrifugal with machinable impellers
- Tube-and-shell heat exchanger
- Purge coil for dissipating excessive solar energy

System components are sized upon available data from the sites, or calculated, or obtained from analysis of system performance. Further refinements in system sizing of pumps and piping can be expected when specific buildings are determined and piping layouts are completed. This could cause considerable variation in the projected size of pumps, pipe and valving.

All valving except control valves are excluded from the transport system. These valve sizes and locations can best be determined after building layouts and will be included in the installation specifications and pricing.

The most critical components on sizing are the heat exchangers. An analysis was made to determine the effects of using various heat exchangers with different "effectiveness" values and mass flowrates. Such an analysis is clearcut for all modes of operation except heating and cooling from storage. In these cases, the transport subsystem consists of two direct-type heat

exchangers coupled by the circulation of a heat-transfer medium. The thermal performance of the subsystem when in these modes of operation was investigated. The storage media and liquid-coupling working fluid were both water. The overall "effectiveness" is related to the subsystem parameters for $C_L > C_s > C_c$ by the following equation:

$$\epsilon = \frac{1}{\frac{1}{\epsilon_c} + \frac{C_c}{C_L}\left(\frac{1}{\epsilon_s}\frac{C_L}{C_s} - 1\right)}$$

For the condition $C_s > C_L > C_c$, the coupling loop-to-storage loop heat-flow capacity ratio, C_L/C_s, drops out. The individual terms of the equation are defined in Figures 4-22 through 4-30.

The results of the thermal performance of the transport subsystem for cooling and heating from storage modes of operation are presented in Figures 4-22 through 4-29. The conclusions of the analysis for the cooling mode in single-family residence parameters' design range are:

- A 12-gpm liquid-coupling loop flowrate has been selected. An increase in flowrate should be accompanied by an increase in storage-loop flow to prevent any decrease in subsystem performance. However, the decrease in subsystem performance is minimal.

- The storage-loop flowrate should be 7 to 9 gpm. Storage flow variation has only a small effect on the overall thermal performance of the subsystem; effectiveness increases 3.5 percent/gpm increase in flowrate.

- The storage-side heat-exchanger effectiveness should be 0.5 to 0.6. A 1 percent overall effectiveness increase can be expected per 5 percent increase in the component heat-exchanger effectiveness.

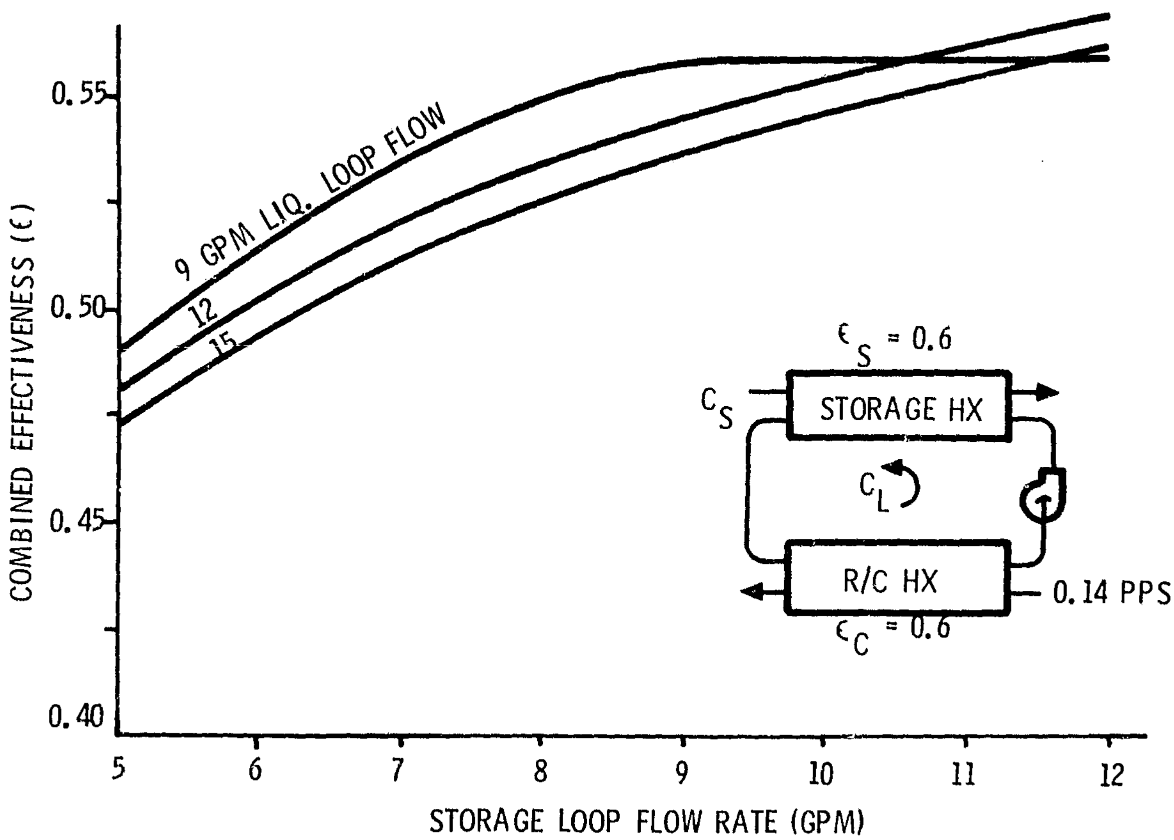

Figure 4-22. Analysis of Cooling Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem

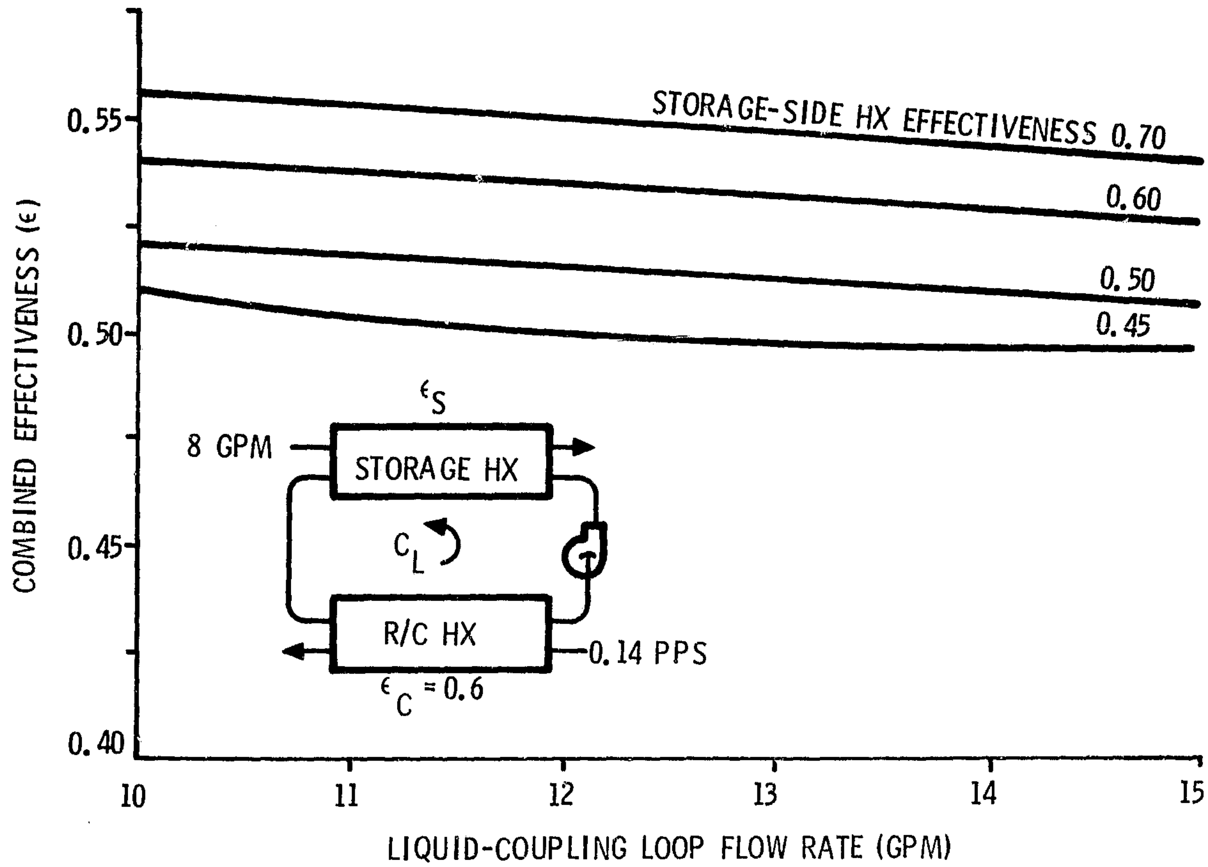

Figure 4-23. Analysis of Cooling Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem

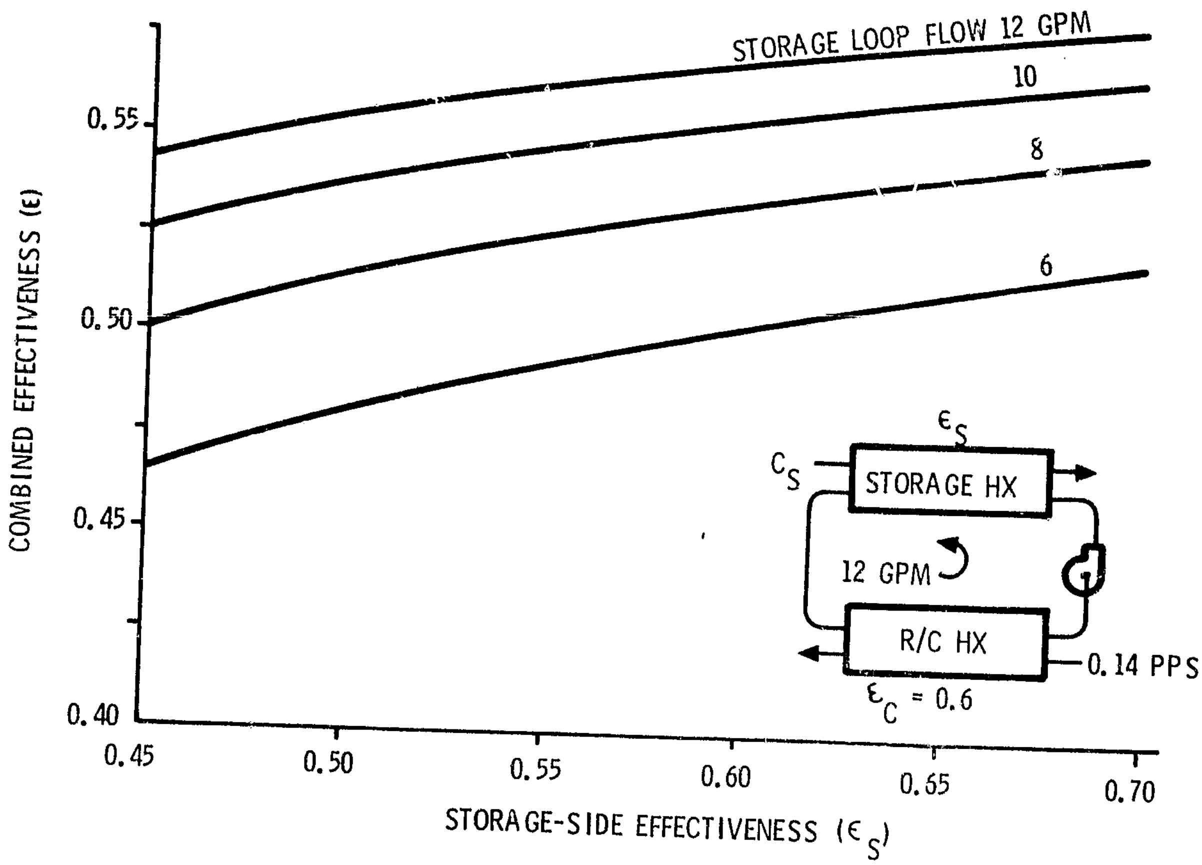

Figure 4-24. Analysis of Cooling Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem

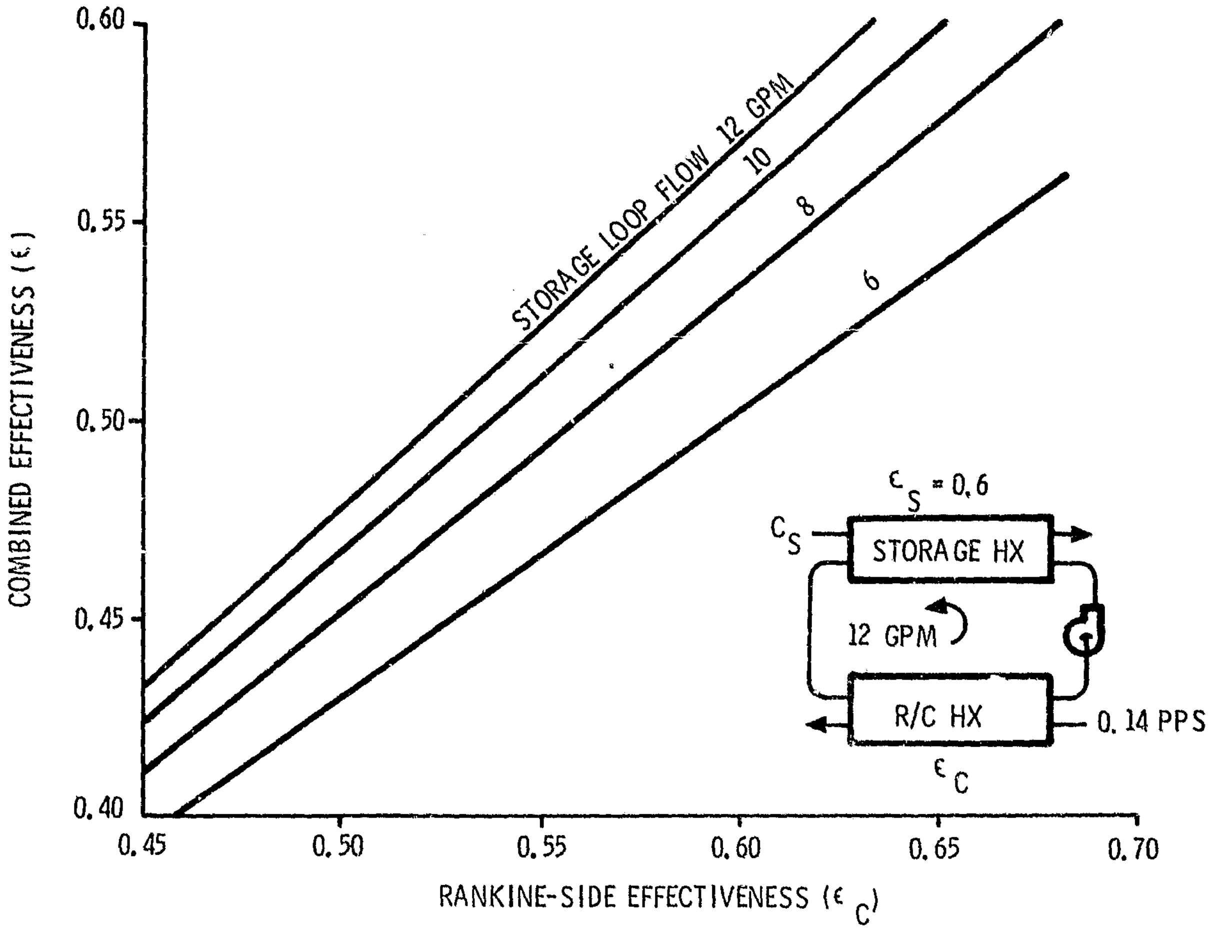

Figure 4-25. Analysis of Cooling Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem

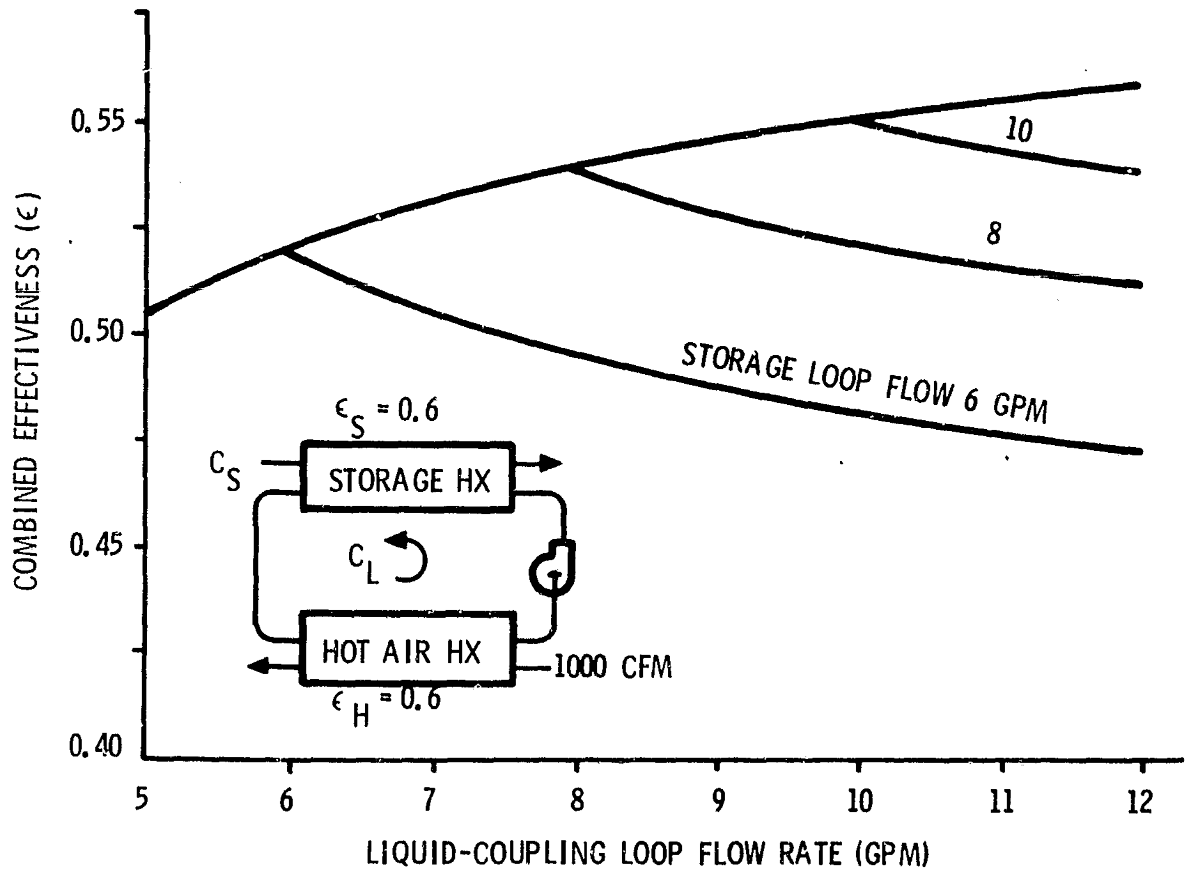

Figure 4-26. Analysis of Heating Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem

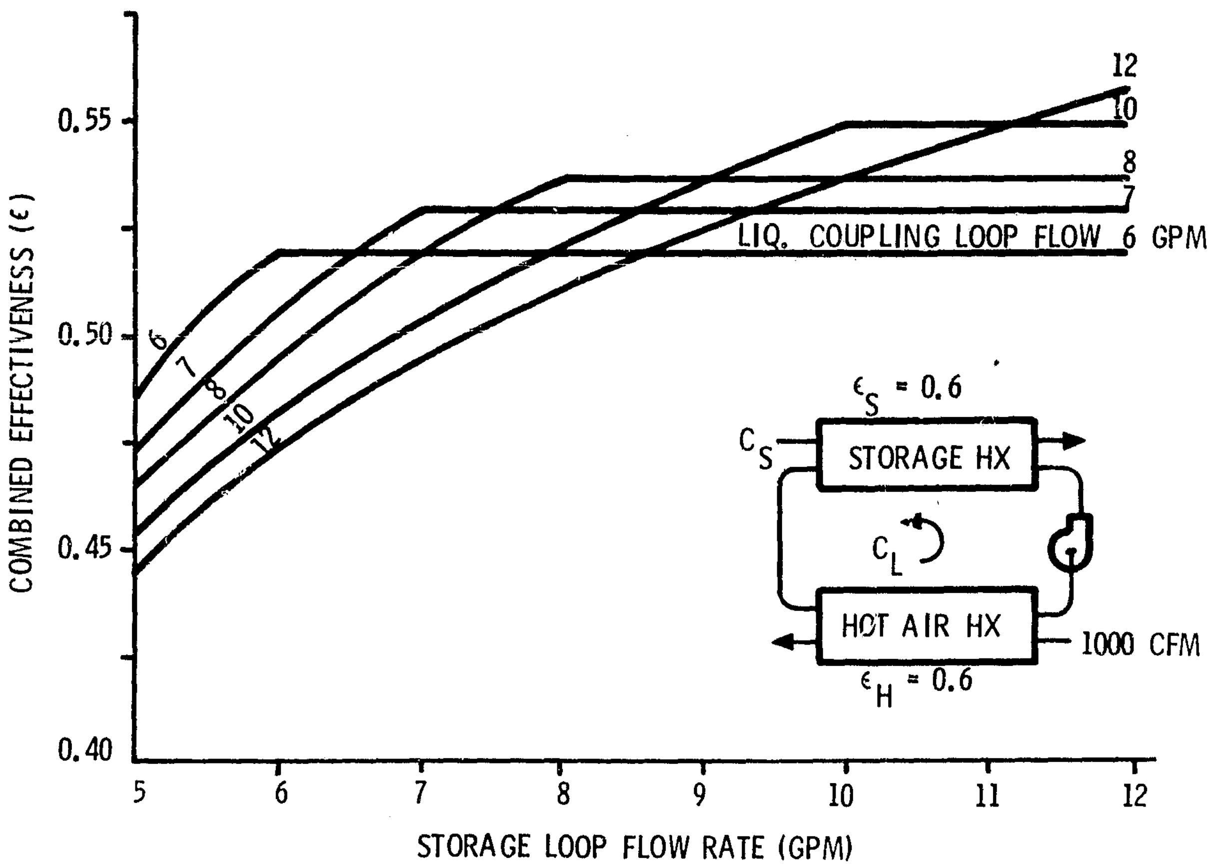

Figure 4-27. Analysis of Heating Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem

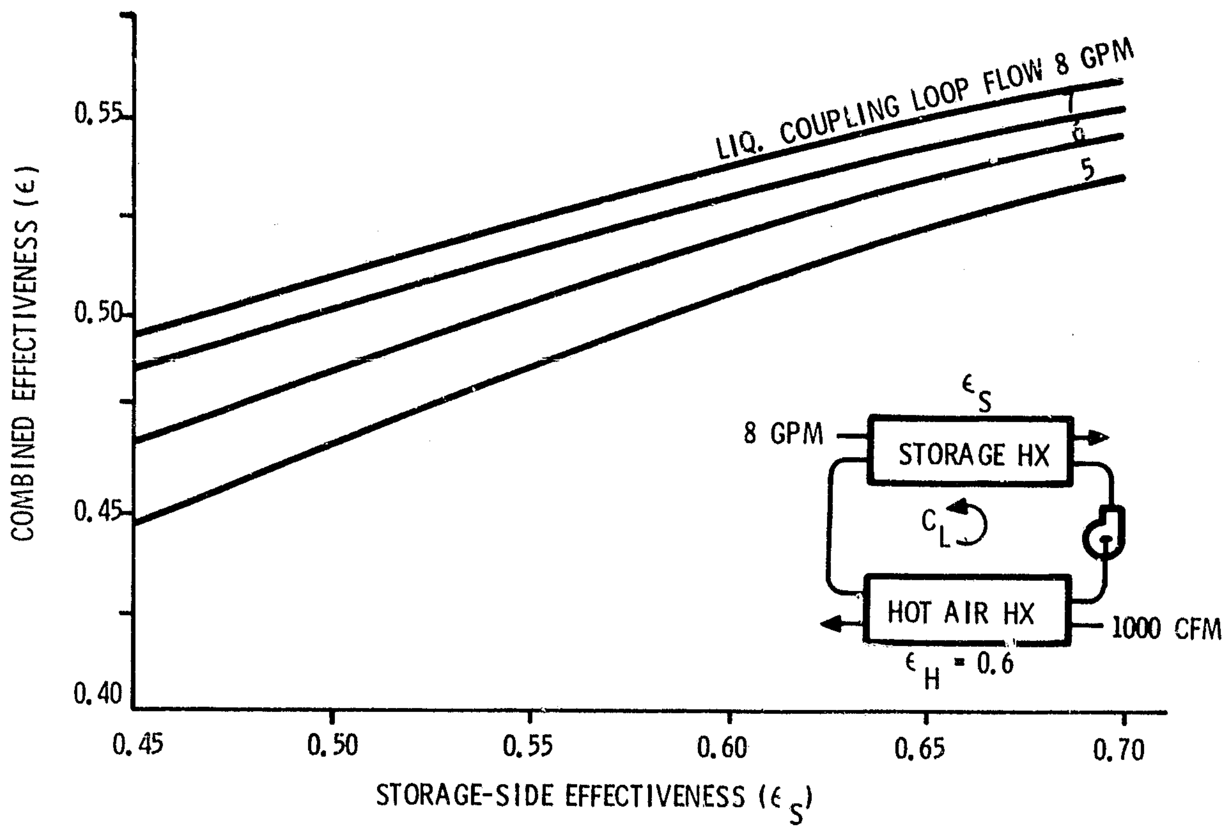

Figure 4-28. Analysis of Heating Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem

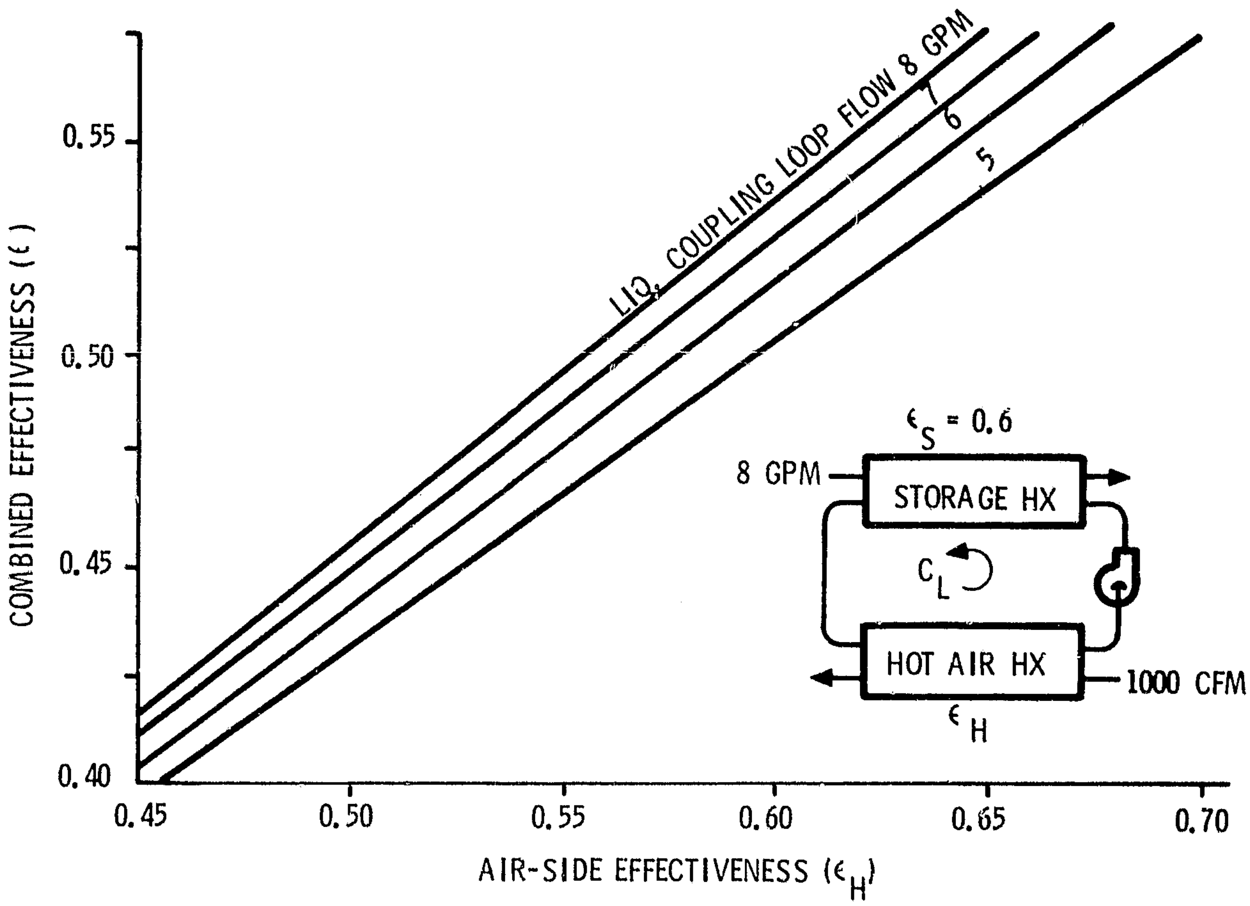

Figure 4-29. Analysis of Heating Cycle Liquid-Coupled, Indirect-Transfer Energy Transport Subsystem

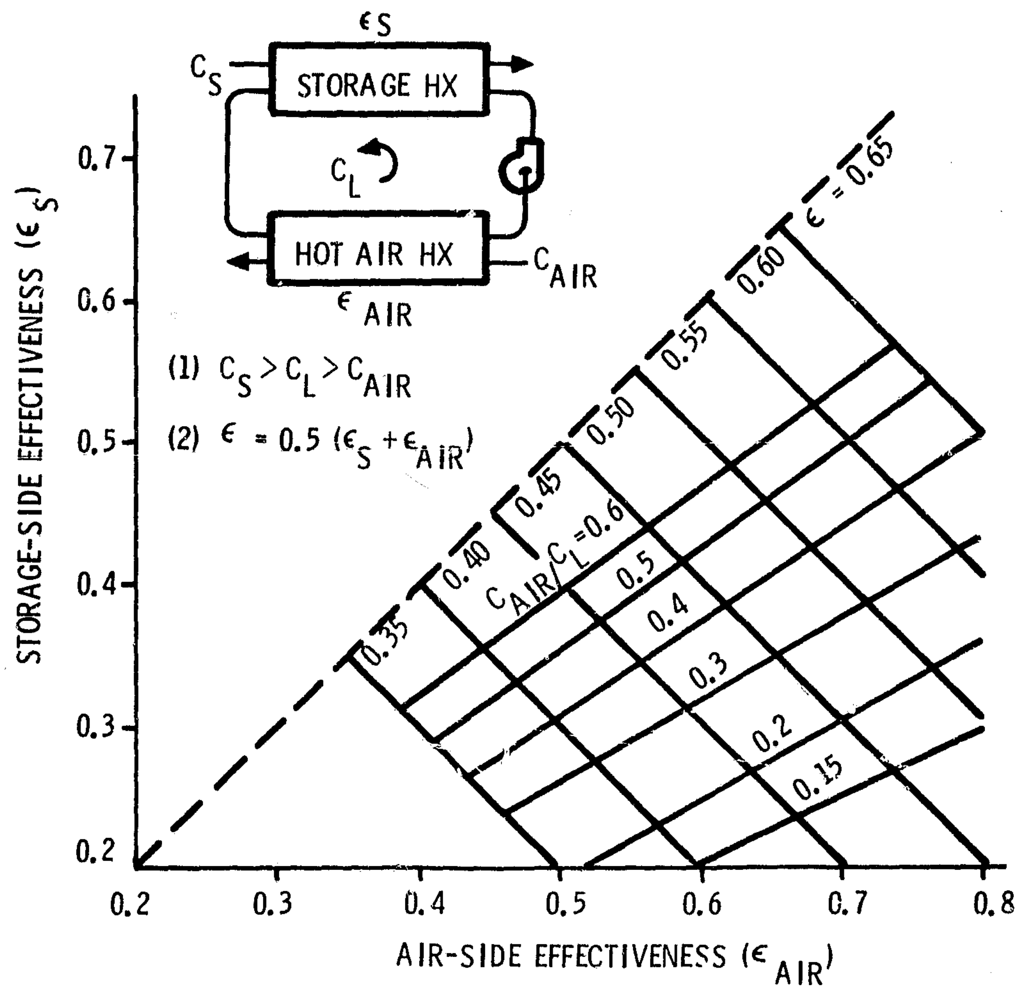

Figure 4-30. Direct-Type Heat-Exchanger Combinations for Maximum Liquid-Coupled Loop Effectiveness

- The cooling-side heat-exchanger performance is the dominant influence on the overall effectiveness of the transport system. An 8 percent overall gain is obtainable per 10 percent improvement in the cooling-side heat-exchanger performance. This is due to the small cooling media flow capacity rate relative to that of the liquid-coupling loop.

The conclusions of the analysis for the heating mode in the parameters' design range are:

- The optimum overall effectiveness occurs when the liquid coupling-to-storage loop heat-flow capacity ratio is unity. However, operation at less than unity ratio values results in an insignificant drop in overall effectiveness.
- The 7- to 9-gpm storage-loop flowrate for cooling also holds for heating purposes. Storage flow variation has a minor effect on the overall thermal performance; effectiveness increases 5 percent/gpm increase in flowrate.
- The liquid-coupling loop flowrate should be 6 to 8 gpm, and variations of flowrate in this range have a negligible effect on the overall subsystem performance. Unlike the cooling mode, increases in flowrate need not be accompanied by a storage-loop flow increase, provided that storage-loop flow is the larger of the two.
- The same conditions exist for the storage-side heat-exchanger effectiveness in the heating mode as exist in the cooling mode.
- The same conditions exist for the air-side heat-exchanger effectiveness in the heating mode as exist for the cooling-side heat exchanger in the cooling mode.

Figures 4-30 and 4-31 describe the results for component heat-exchanger optimization for the heating-from-storage mode of operation. Figure 4-30 gives component heat exchanger combinations which optimize the subsystem overall effectiveness. The constriction which must be satisfied is that only heat exchanger combinations with equal average effectiveness magnitudes can be compared [i.e., equal values of $\epsilon = 0.5 (\epsilon_s + \epsilon_{air})$]. Figure 4-31 gives the overall subsystem effectiveness for the conditions of Figure 4-30. These results provide a basis for thermal performance and heat-exchanger combination cost analysis.

Three working-fluid candidates for the baseline heating/cooling solar systems were compared. The fluids were 50 percent aqueous solutions of Dowtherm-J (alkylated aromatic liquid), Dowfrost (propylene glycol), and Dowtherm SR-1 (ethylene glycol). General properties and cost of each fluid, and their performance relative to water were evaluated. The performance constriction was that all fluids produce an equal heat-transfer effect. Table 4-1 gives a cost comparison for the three fluids. Figure 4-32 and Table 4-2 shows fluid performance over the temperature range 100° to 215°F. Based on these results alone, Dowtherm-J was dismissed from further consideration. Dowtherm SR-1 shows somewhat better performance than Dowfrost; however, Dowfrost is somewhat less expensive.

Table 4-1. Cost Comparison of Dowtherm-J, Dowfrost, and Dowtherm SR-1 in Dollars/Gallon

Quantity	Dowtherm-J	Dowfrost	Dowtherm SR-1
4000 gal	$3.25	$2.70	$2.90
70 drums	3.38	3.14	3.37
10 drums	3.87	3.31	3.56

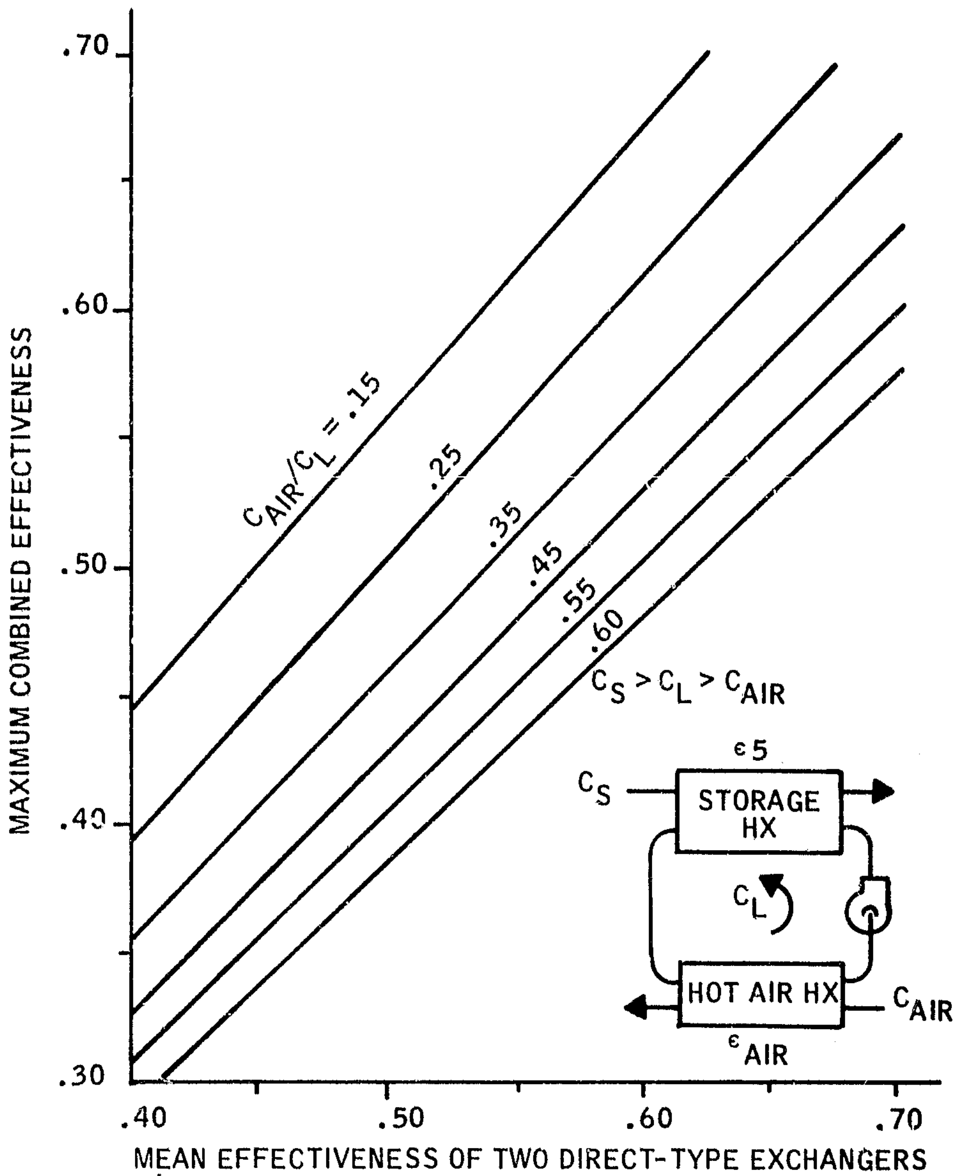

Figure 4-31. Maximum Effectiveness of Liquid-Coupled, Dual-Exchanger Heat Transport Loop

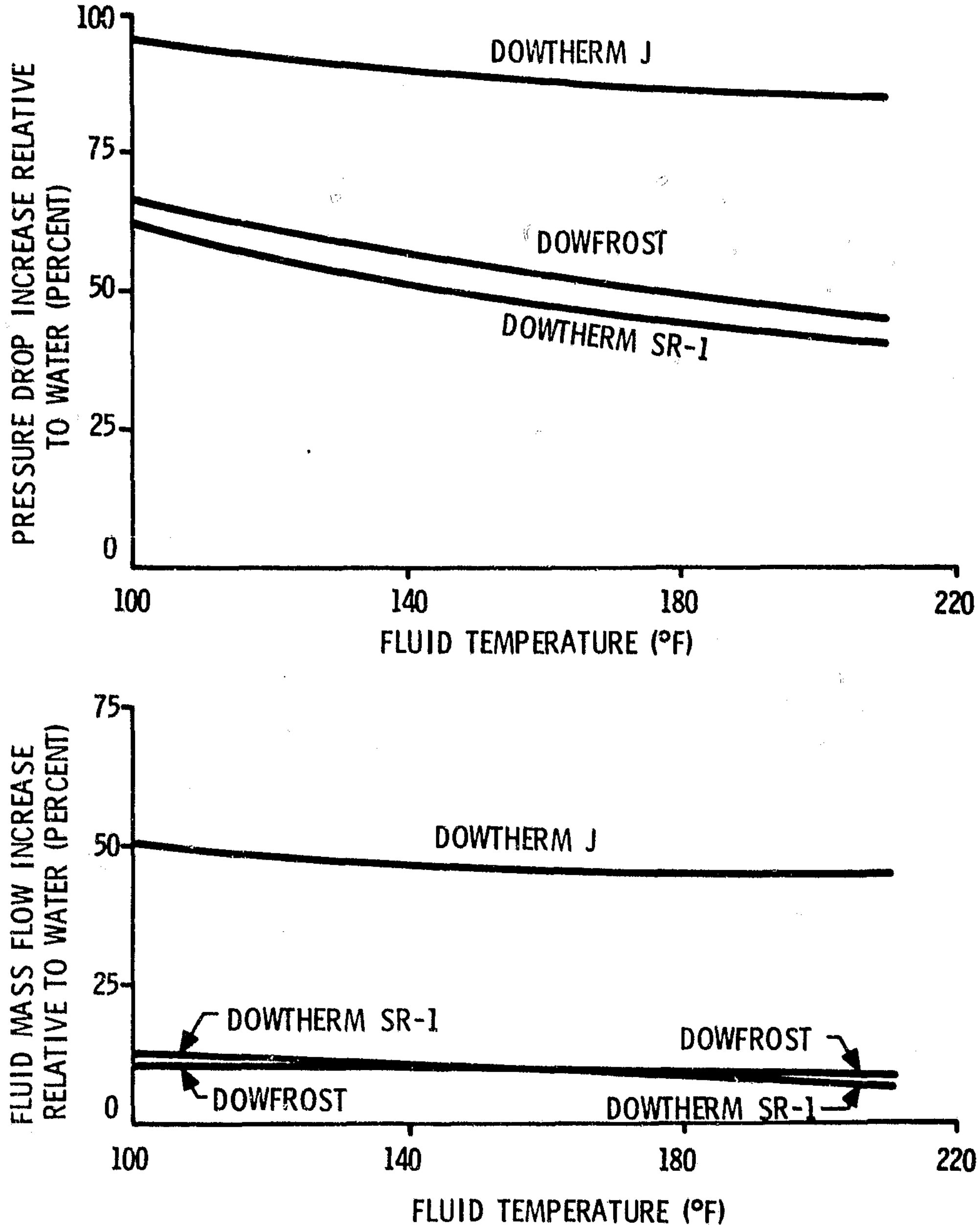

Figure 4-32. Performance of 50-Percent Aqueous Solution Relative to Water for Condition of Equal Heat Transport

Table 4-2. Working Fluid Performance with Equal Heat Transport Criteria

Parameter	Water			Dowtherm-J			Dowfrost (propylene glycol)			Dowtherm SR-1 (ethylene glycol base)		
Dilution (% by volume)				50			50			50		
Pressure (psia)	14.7			14.7			14.7			14.7		
Freeze point (°F)	32			-100°F for pure sol.			-28			-36		
Boil point (°F)	212			575°F for pure sol.			214			230		
Temperature (°F)	100	140	180	100	140	180	100	140	180	100	140	180
Density (lbm/ft^2)	62.0	61.4	60.6	57.6	56.7	55.8	64.1	63.0	62.0	66.2	65.3	64.4
Specific heat (B/lbm-°F)	0.997	0.998	1.002	0.724	0.733	0.744	0.870	0.885	0.900	0.830	0.850	0.870
Kinematic viscosity (ft^2/hr)	0.0266	0.0186	0.0139	0.0284	0.0200	0.0152	0.121	0.0625	0.0423	0.100	0.0625	0.0423
Heat capacity Rate ratio	1.0	1.0	1.0	0.665	0.678	0.685	0.903	0.910	0.919	0.889	0.907	0.923
Flow capacity ratio for same heat transfer	1.0	1.0	1.0	1.505	1.476	1.461	1.108	1.099	1.088	1.125	1.103	1.083
Reynolds number ratio relative to water	1.0	1.0	1.0	1.409	1.373	1.335	0.244	0.327	0.358	0.299	0.328	0.356
Friction factor ratio relative to water	1.0	1.0	1.0	0.942	0.946	0.951	1.280	1.216	1.197	1.235	1.215	1.198
Pressure drop ratio relative to water	1.0	1.0	1.0	1.952	1.902	1.870	1.625	1.507	1.450	1.669	1.573	1.485

In general, in addition to low-temperature freeze protection and high boiling point, Dowfrost and Dowtherm SR-1 exhibit numerous similar characteristics. Some of these are:

- Stability over wide temperature range
- Low coefficient of thermal expansion. If the aqueous-glycol solution undergoes complete freeze-up, upon thawing it will not crack the container.
- Noncorrosivity with inhibitor additives
- Completely nonflammable when above 20 percent aqueous solution
- Extremely low toxicity
- Annual check of the solution inhibitor level is normally required.

It should be noted that Dowfrost and its inhibitor meet the requirements of the Food Additives Regulation which lists those materials "generally recognized as safe" for use in foods.

Aqueous solutions of both Dowfrost and Dowtherm SR-1 appear to be capable working fluids for the baseline heating/cooling solar system. However, before a final decision is made, several other prospects, such as silicone oil, shall be investigated.

4.9 CONTROLS SUBSYSTEM

The control subsystem design is based on the overall system definition that includes related subsystems such as collectors, solar storage, energy transport, and auxiliary heating/cooling. Control logic is determined so that solar energy is collected and used to minimize the consumption of conventional energy while maintaining conditions in the space consistent with the type and duration of occupancy. Through engineering analysis, operating modes that could be hazardous or lead to discomfort are eliminated. The heating or cooling system demand for energy is satisfied by using solar energy before adding conventional energy required to satisfy the load. Interrelated subsystems that affect the operating efficiency of other subsystems are also analyzed. This analysis determines a control logic that leads to a higher overall system efficiency. For example, if stored energy is higher than collected energy, the control subsystem prevents dissipation of the stored energy into the collector subsystem.

The control logic is used to design a control schematic of the subsystem and to determine the operating temperatures, pressure, flow, and differential-temperature requirements of the system. This information is then used to determine component requirements. The component requirements are then used to establish availability of off-the-shelf deliverables and items requiring complete component development. Off-the-shelf deliverable controls would include standard thermostats, control valves, motor operators and linkages, switching and time-delay relays, time clocks, etc. These are all standard components as used in the HVAC industry.

The differential-temperature controller is also a standard production item. This controller, when used with the proper thermistor sensors, is designed as a module capable of providing a variety of automatic control functions in the switching of circulating pumps, valves, dampers, motors, and other accessories used in solar control systems. It has a solid-state differential

amplifier with a three-pole switching relay. Control functions can be modified by changing the connections of the differential resistors, setpoint resistors, and thermistor sensors. This module can be used as a:

- Differential-temperature control (relay makes on temperature differential rise)
- Setpoint temperature control (relay makes on temperature rise)
- Setpoint temperature control (relay makes on temperature drop)

The differential-temperature controller has the following design features:

- Modular construction (one basic module provides a variety of solar control functions)
- Solid-state differential amplifier
- Three-pole switching relay (two N.O. and one N.C. isolated contacts)
- Integral transformer for powering the low-voltage control circuit
- Color-coded leadwires for line-voltage connections
- Exposed terminal strip with screw terminals for low-voltage connections
- Plug-in differential resistors
- Mounts in any position on a standard 4 x 4-inch junction box
- Interchangeable thermistor sensors

Control equipment requiring complete component development would include the control panel and the motor control panel.

The logic control panel will provide the control logic needed to efficiently and safely operate the solar-energy collection, transport, and auxiliary-energy subsystems. This is more complex than conventional systems of similar size and application. For this reason, the control logic is contained on a printed-circuit card using multipole relays or solid-state devices whose control and interconnection requirements are pre-engineered and prewired. This allows high reliability and contributes to simplification of the installation and maintenance of the control subsystem. To accomplish this, a logic control panel (panels) will be developed and tested. It is anticipated that two unique logic panels will be required, one for the heating-only subsystem and one for the heating and cooling subsystem. The additional logic specific to single-family, multiple-family or commercial applications will be addressed in the master control panel discussed below.

The master control panel will also require complete component development. As stated above, the control system requirements are more complex for combined solar-energy and conventional-energy systems than are those for conventional systems alone. In order to ease the installation and maintenance problems, a master control panel for each unique control system will be developed. It is anticipated that all components of the control subsystem not required to be remotely located to sense temperature or control flow will be enclosed in the master control panel. This includes the differential-temperature controller, logic control panel, time clock, time-delay relays, and power relays. Electrical connections to remote sensors and actuators will be easily identified via terminal connections. The use of the master control panel also allows a control subsystem to be fabricated, prewired, pretested, and completely checked out during acceptance and before delivery to installation sites. By using standard components in addition to the logic control panel controller, six unique master control panels will be developed. They are: 1) heating-only single-family; 2) heating-only multiple-family; 3) heating-only commercial; 4) heating/cooling single-family; 5) heating/cooling multiple-family; and 6) heating/cooling commercial. As site specific data become available, these control panels will be developed.

4.9.1 Development and Qualification Tests

During development, the components listed above will be tested at the bread-board level using complete simulation of each operational mode for the system. In addition, failure modes, such as power restoration after power loss, will be tested to ensure that no damage or unsafe condition that can be reasonably prevented will occur due to control subsystem malfunction. In addition, tests will be run to determine the best human-engineering factor for the control system, with the goal being to have a system requiring no more "operator expertise" than presently required for a conventional system of similar size and application. The IPC and other applicable criteria will be used to determine development and performance requirements.

At the completion of breadboard testing, printed-circuit board testing of the logic controller will be done to assure design and development accuracy. No tests are anticipated for the control subsystems during the qualification phase, as components used during development will be standard manufacturer's catalog items tested during development, or tested in the acceptance phase as discussed under acceptance testing.

4.9.2 Acceptance Testing

Each of the 12 master control panels will be tested using all system operation modes as described for the heating and heating/cooling subsystems. This includes tests of the logic controller and differential-temperature controller. Remote thermostat sensors and control valves are mass-produced standard catalog items used in large quantities in similar applications and service and will not be tested prior to installation.

4.10 SITE DATA COLLECTION SUBSYSTEM

The preparation of the detailed flow schematic used to identify the instrumentation desired will be completed shortly after the specific sites are identified. Typically, this effort will be done in parallel with the preparation of the installation drawings. The evaluation factors listed in Table 3-1 of referenced SHC-1006 will be used when preparing this instrumentation-oriented schematic. The, specific sensors will be selected from the Acceptable Sensor List, reference Table 4-2 of SHC-1006, and a Prepared Instrumentation Plan prepared. This plan will be site-specific (i.e., six plans for heating/cooling systems) dependent. When approved (i.e., AIP), it will be forwarded for the site owner/architect. Following that, the site owner/architect will prepare the detailed construction drawings, and, using these, the HVAC contractor will provide the installation costs. A modified list of action "steps" shown in Table 1-1 of SHC-1006 is included herein as Table 4-3.

Table 4-3. Instrumenting Steps*

Step	Action by	Action
1	NASA	Supplies "instrumentation installation guidelines" to Honeywell.
1A	NASA	NASA/MSFC supplies sites and site data.
2	Honeywell	Prepares complete solar system schematics showing instrumentation locations for each site.
3	Honeywell	Submits Proposed Instrumentation Plan (PIP) to NASA for review and approval (without costs).
4	NASA	Reviews and modifies PIP as necessary. Supplies Honeywell with an Approved Instrumentation Plan (AIP)
4A	Honeywell	Provides site owner/architect with AIP.
4B	Architect	Provides Honeywell with detailed construction drawing.
4C	Honeywell	Obtains costs to install from HVAC contractor.
5	NASA	Negotiates a contract modification with Honeywell to implement the AIP.
6	NASA	Provides and ships all necessary instrumentation as Government-Furnished Equipment (GFE) to the demonstration site for HVAC contractor installation and checkout.
7	NASA	Arranges for the installation of the necessary telephone transmission equipment to accommodate the Site Data Acquisition Subsystem (SDAS).
8	Honeywell	Informs ERDA that the AIP has been implemented and that all instruments have satisfactorily undergone checkout.
9	NASA/ ERDA	Delivers and installs an SDAS and confirms the satisfactory transmission of the data to the Central Data Processing Facility.
10	ERDA/ NASA	Receives and evaluates the incoming performance data, and prepares monthly and annual performance reports using standardized data reduction procedures.
11	NASA	Mails performance reports for each site to Honeywell within 30 days after the end of each performance period.
12	ERDA/ NASA	Removes SDAS from the site at the end of the test period.
13	Honeywell	Upon NASA's direction, removes all instrumentation which can be cost-effectively removed and returns to NASA at the end of the test period.

*Reference SHC-1006, Table 1-1: Procedure for Instrumenting Solar Heating and Cooling Commercial Demonstration Projects.

4.11 ELECTRICAL SUBSYSTEM

The electrical subsystem consists of those components necessary to deliver power to the subsystems. The wiring, conduit, and installation hardware are best determined after specific buildings have been identified. At that time, complete drawings and specifications can be completed. The components of this subsystem which can be identified and can be designed for electrical requirements are:

- Cooling tower fan motor starter
- Purge coil fan motor starter
- Magnetic motor starters for pumps
- Fused disconnect switches
- Circuit breakers
- Air handler

Equipment sizing is based on National Electrical Manufacturer's Association (NEMA) standards, and all items purchased will be in accordance with Underwriter's Laboratory requirements and will bear the UL label of acceptance.

4.12 COOLING SUBSYSTEM

4.12.1 Cooling Subsystem Description

4.12.1.1 General -- Systems developed for this project will be new in design, much different from any product that Lennox currently produces. The equipment includes two sizes, one to meet the needs of the residential market and one to meet the needs of the multiple-family/commercial market.

The model codes for the residential and the multiple-family/commercial units are RSH1 (Rankine-powered high side) and RCWS1 (Rankine-powered chilled-

water system). The code letters and series designation will be followed by the size, phase, dash number, and voltage designation. The complete model numbers are as follows:

- Residential unit - RHS1-411V-1P
- Multiple-family/Commercial unit - RCWS1-3053V-1Y

4.12.1.2 Interface -- Under present agreement, Lennox is to supply the refrigeration subsystems, sheet-metal cabinetry, and structural framework containing both the refrigeration subsystem and the Rankine-cycle engine. Barber-Nichols Engineering will develop and supply the Rankine-cycle engines.

Both residential (Figure 4-33) and multiple-family/commercial (Figure 4-34) refrigeration subsystems use open-type compressors and water-cooled condensers. The residential unit is a "split system" by category and is linked to a direct-expansion evaporator coil. The multiple-family/commercial unit is "packaged" by category and incorporates a liquid chiller as the evaporator to supply chilled water to the conditioned space.

4.12.1.3 Operation -- The cooling subsystem consists of two distinct thermodynamic portions, the Rankine-cycle (R/C) power loop and the air-conditioning (A/C) loop. The combined RC/AC subsystem is shown schematically in Figure 4-35.

In the Rankine cycle, working fluid is pumped from a water-cooled condenser through a regenerator to the boiler, which extracts heat from solar collector water. The regenerator is a liquid-to-vapor performance improvement heat exchanger operating within the R/C loop. Vapor leaving the boiler is admitted to nozzles which feed a turbine rotor. Turbine exhaust vapor passes through the vapor side of the regenerator and returns to the condenser, completing the Rankine cycle.

Turbine rotational speed is reduced by a gearbox whose low-speed shaft is connected by an overrunning clutch to a motor-generator and air-conditioning compressor. This configuration permits total input power to the A/C loop from the solar-powered R/C. If the R/C system cannot keep up with the cooling demand (i.e., if it is a partially cloudy day), rated cooling can be maintained with the help of the motor.

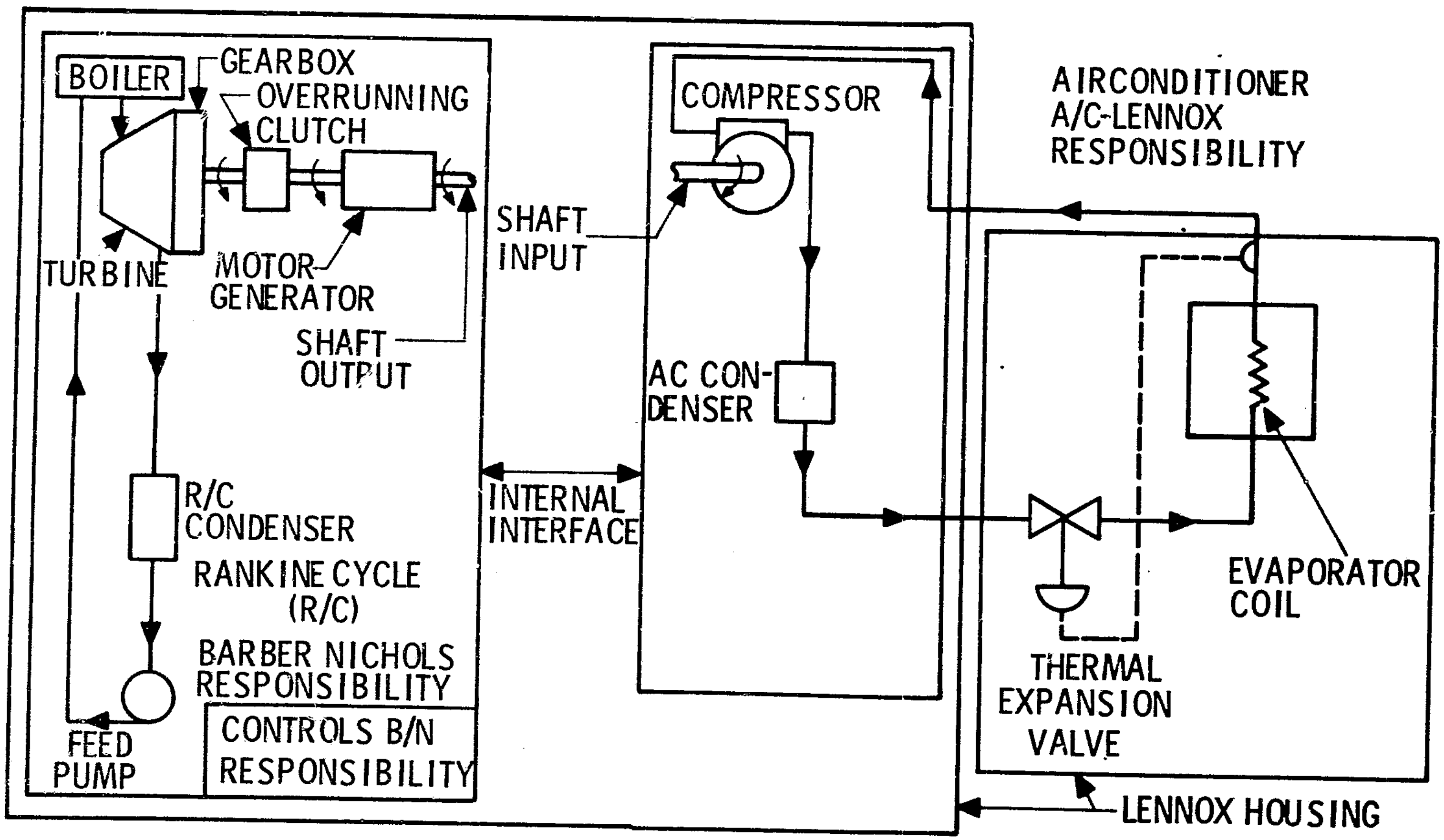

Figure 4-33. Residential Cooling Subsystem Interface

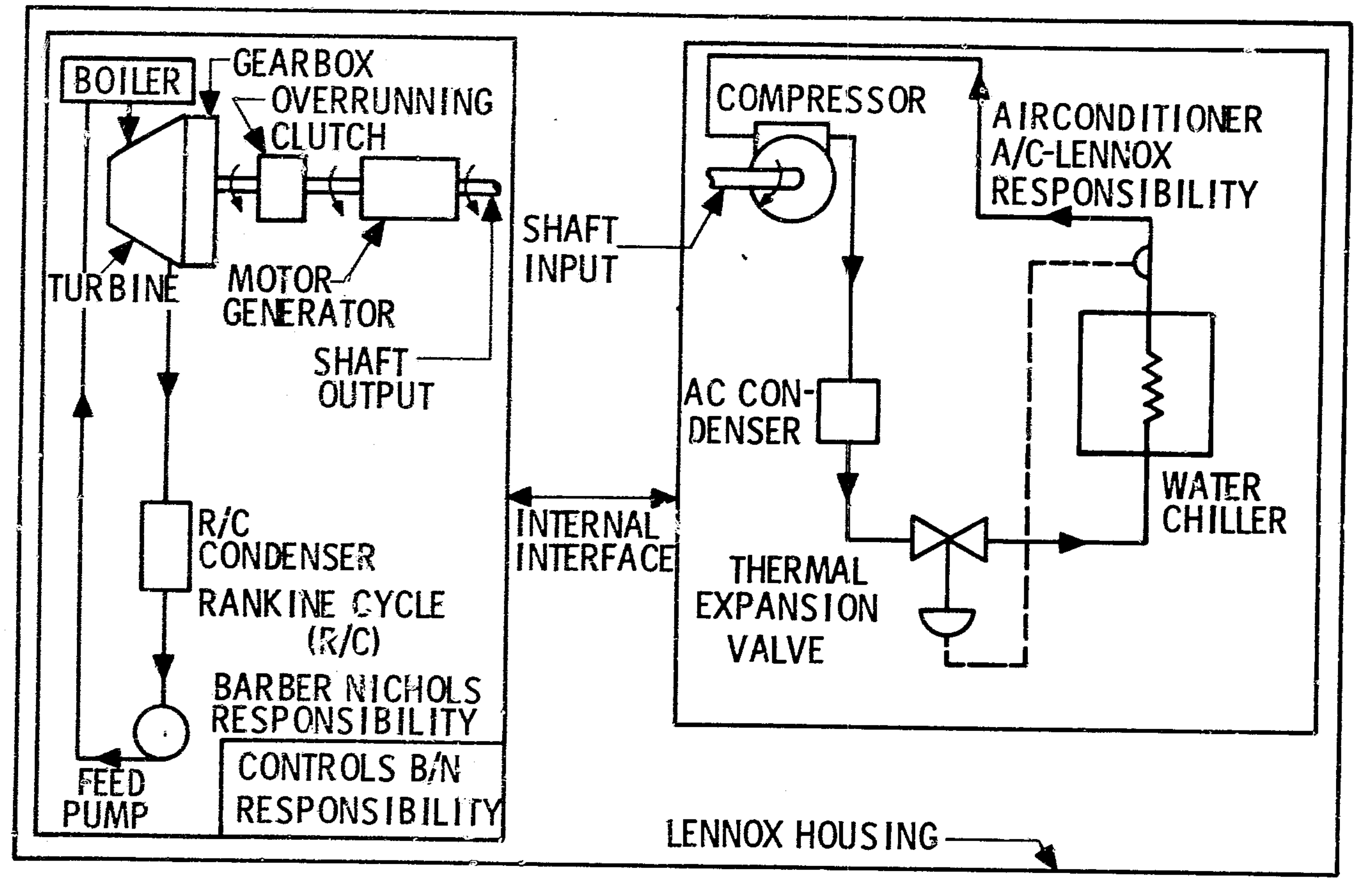

Figure 4-34. Multiple-Family/Commercial Cooling Subsystem Interface

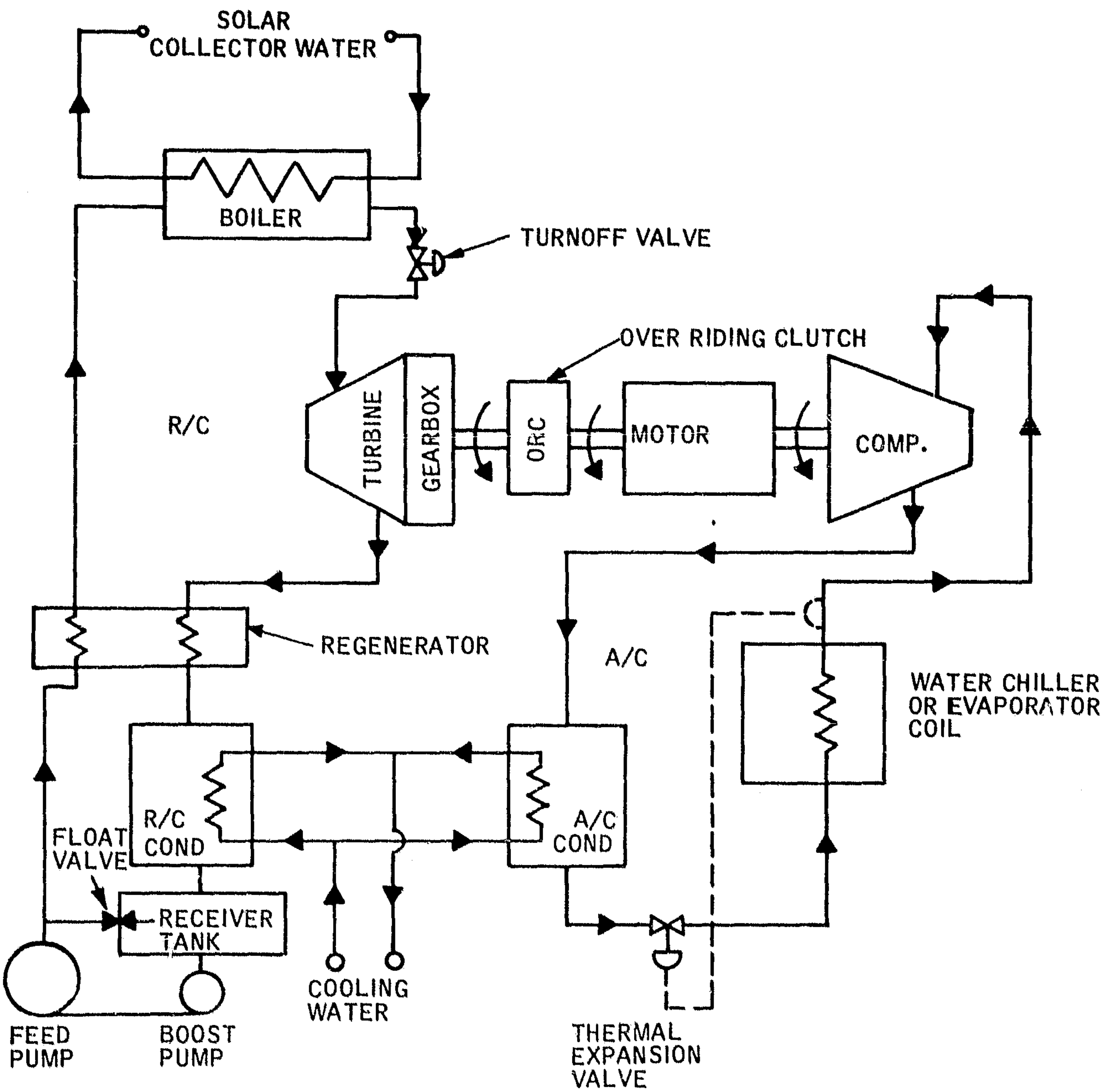

Figure 4-35. General System Schematic for Rankine-Cycle Air-Conditioning System

In the air-conditioning cycle, a compressor receives low-pressure refrigerant vapor from the evaporator (or chiller) and pressurizes it. The high-pressure vapor then enters a water-cooled condenser where the latent heat of vaporization is removed, leaving high-pressure liquid refrigerant. This liquid is allowed to pass through a thermal expansion valve to the low-pressure portion of the loop (i.e., evaporator). The expansion of this high-pressure liquid produces a mixture of refrigerant liquid droplets and vapor at a low temperature (about 45°F). This low temperature provides sufficient temperature difference to extract heat from air or water, as desired, and thus supply cooling. The energy taken from the air or water in the cooling process supplies heat of vaporization to the droplets of refrigerant liquid, producing refrigerant vapor. This low-temperature, low-pressure vapor then flows to the compressor, completing the A/C cycle.

4.12.2 Refrigeration Design Approach

The Lennox approach to the NASA-404 Program is to develop a product which will economically bind together energy efficient hardware, consumer safety, producibility, and marketability in a common package. An economical design will be achieved by striving to select those components which are in common use in today's industry. The method by which these components are applied will differ from current designs, however.

The skin for the cabinetry of both the residential unit and the multiple-family commercial unit will be an adaptation of cabinetry currently used to contain Lennox CHA9 and DSS1 product lines. The present cabinets will be dimensionally modified to meet the package requirements of the 50 inches wide by 40 inches high by 56 inches long residential unit, and the 84 inches wide by 65 inches high by 120 inches long multiple-family commercial unit. Figure 4-36 is an isometric view of the residential unit.

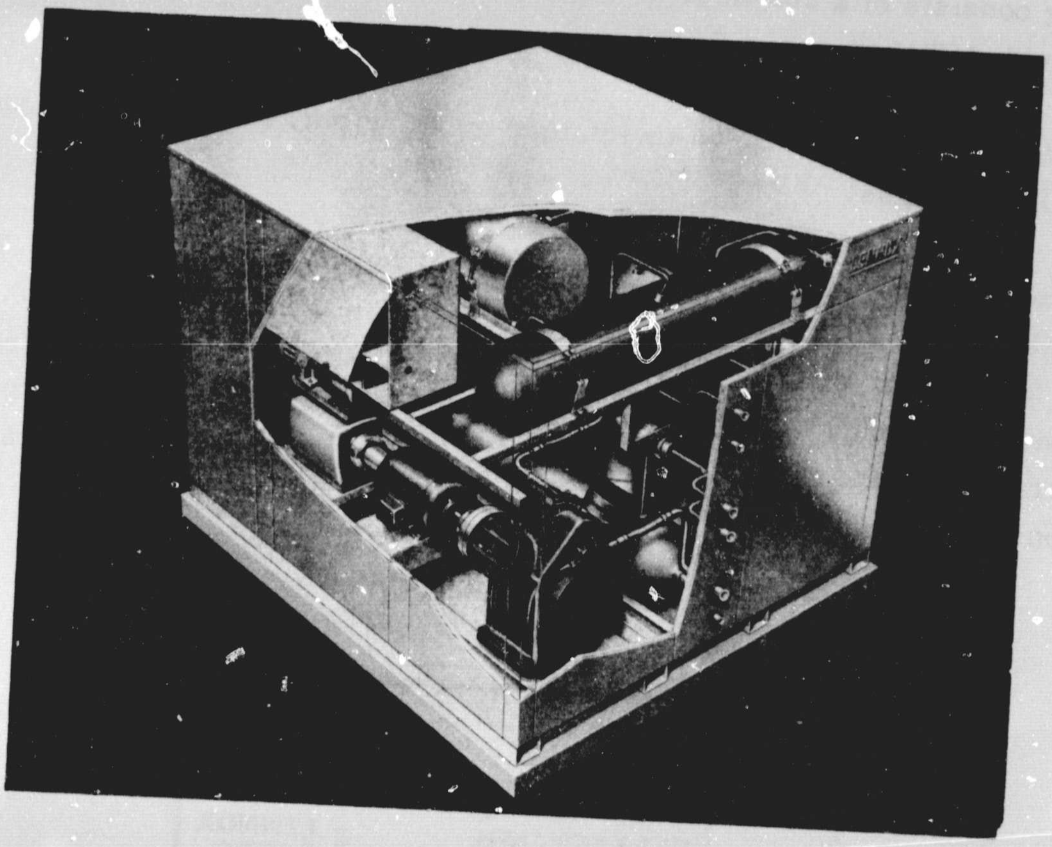

Figure 4-36. Isometric View of Residential Refrigeration Unit

The units will be painted with Lennox industrial-grade acrylic enamel finish. This finish meets the Lennox specifications, which have been developed in accordance with American Society for Testing Material (ASTM) Standard D-1654 (Evaluation of painted or coated specimens subject to corrosive environments) and Standard D-714 (Evaluating degree of blistering of paints). Testing consists of a 500-hour salt spray test, which is conducted in accordance with ASTM Standard B-117.

Marketable aesthetics will be maintained by using current Lennox silhouette and color schemes.

Manufacturability will be achieved by using the concept of modularity. This concept will use advanced subassembly of several modules that will bolt and braze together to form the final package. This will be a vital format for economical manufacturability. The Lennox refrigeration module and the Barber-Nichols heat-exchange module and power module occupy separate sections of the cabinet as shown in Figure 4-37.

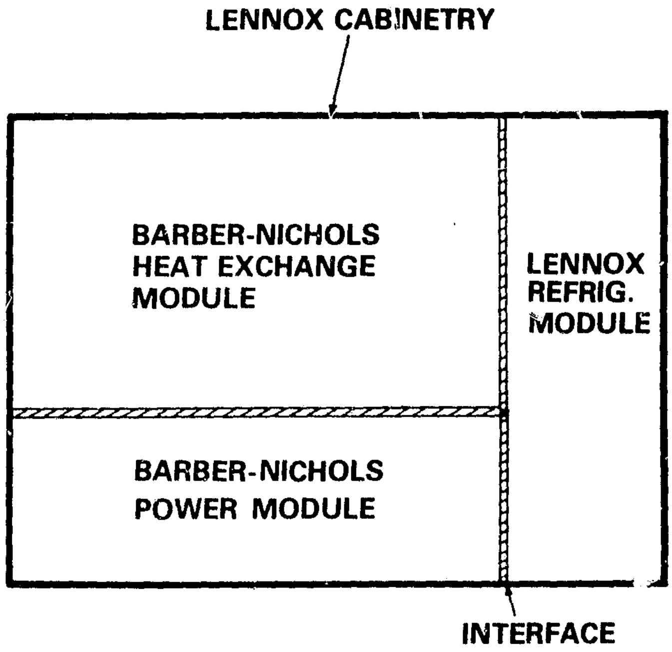

Figure 4-37. Plan View of Residential Unit Modularity

This independence allows Lennox to assemble the refrigeration module into the cabinet for shipment to Barber-Nichols for final assembly and test. Independent module tests can be performed during assembly to determine performance prior to final assembly.

Both the residential unit and the multiple-family commercial unit will be designed for ease of installation and serviceability. Lennox feels that if the installer and servicemen's job is made easier, he will then do a better job during the initial installation and a more thorough job maintaining the system.

Field connections will be centered in one location of the unit. All plumbing connections will be made in this location, external to the cabinetry. This central location will include large access panels to facilitate ease of entry to moving mechanical components and electrical controls as shown in Figure 4-38.

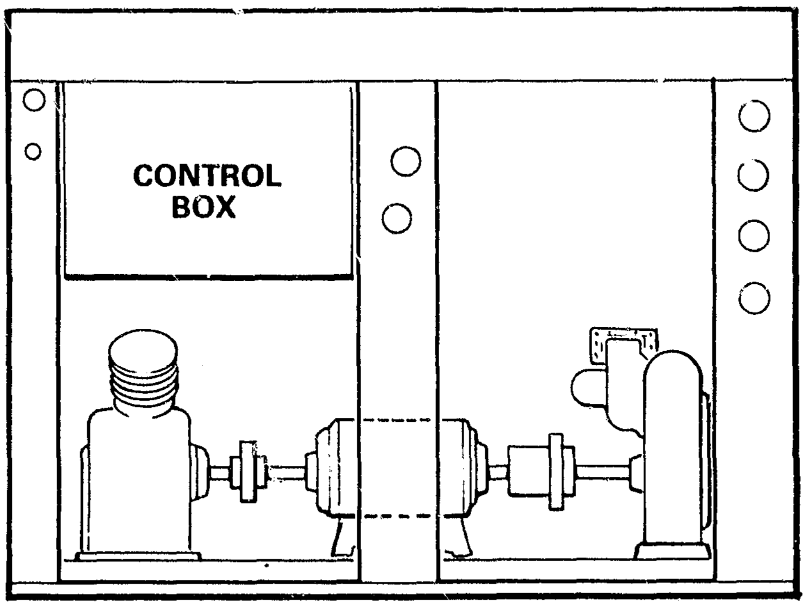

Figure 4-38. Multiple-Family/Commercial Unit Serviceability (Access Panels Removed)

4.12.2.1 Component Hardware -- Lennox will make every attempt to make this program as practical as possible. Component selection will incorporate the off-the-shelf concept in its design whenever possible. By using this concept, Lennox feels that an extremely practical cost-effective system will result. An energy-efficient system will result by applying standard components in an unconventional manner.

Lennox is researching possible compressor options available in the marketplace. The present options being examined are (Figure 4-39).

- Automotive (reciprocating and swash plate)
- Reciprocating
- Centrifugal
- Rotary

Elements of concern for compressor selection include:

- Capacity
- Brake horsepower input
- Coefficient of performance
- Input speed
- Lubrication
- Reliability
- Physical size and weight
- Refrigerant type
- Cost

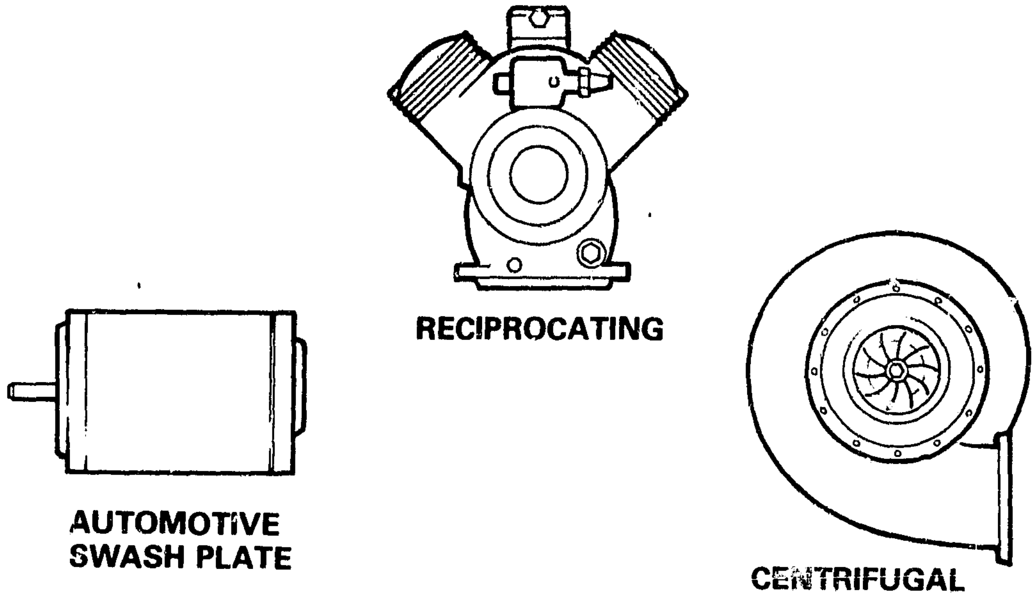

Figure 4-39. Compressor Options

Each of these items is under consideration in a time study scheduled for completion 15 January 1977. Results of the study will be presented at the Preliminary Design Review (PDR). These results determine the final interface between the compressor, motor, gearbox, and turbine. Horsepower has been selected. The remaining critical parameter is rpm, which depends on the type of compressor. It is desirable that this rpm be in a range compatible with a motor synchronous speed (1700 or 3600 rpm) so that the direct drive baseline can be maintained.

Shaft alignment and coupling interface will be carefully coordinated between Lennox and Barber-Nichols (Figure 4-39). Present plans call for rigid mounting of the Rankine-power module and refrigeration compressor on a set of common rails. The rails will be fixed to the unit base by spring mounts. This will allow the rigid assembly to be free floating and hence eliminate any extreme amounts of vibration from being transferred the main body of the unit.

In addition to the spring mounts, flexible connections will be included in all plumbing lines that are interfaced between the Rankine-power module, the refrigeration compressor, and the remaining unit modules as shown in Figure 4-40.

Detailed analysis of evaporator and coil considerations are in process and the status of these trades will also be presented at the PDR.

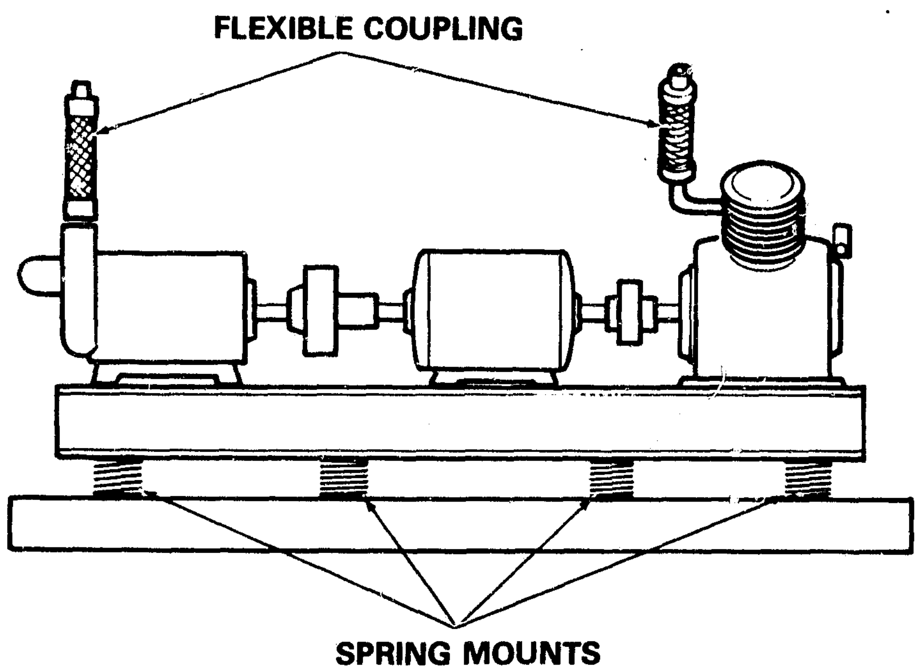

Figure 4-40. Shaft Alignment and Vibration Isolation

4.12.2.2 Controls --

- Residential Refrigeration Module (Figure 4-41) -- Standard controls to be included in the residential refrigeration module are:

  - Electrical: High-pressure switch, low-pressure switch, oil safety switch, crankcase heater and relay (outdoor installations)

- Multiple-family/Commercial Refrigeration Module (Figure 4-42) -- Standard controls to be included in the multifamily-commercial refrigeration module are:

  - Mechanical: Thermostatic expansion valve, compressor capacity control

  - Electrical: High-pressure switch, low-pressure switch, oil safety switch, crankcase heater and relay (outdoor installations) Freeze stat.

Lennox will, whenever possible, use control components currently applied in Lennox equipment to assure component reliability.

4.12.2.3 Refrigeration Module Performance -- Present design goals call for the following performance characteristics for the refrigeration modules:

- Residential Subsystem:
  - Capacity - 36,000 Btuh
  - Bhp Input - 2.4 Bhp
  - COP - 6.0*

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* Per ARI Standard 210-75

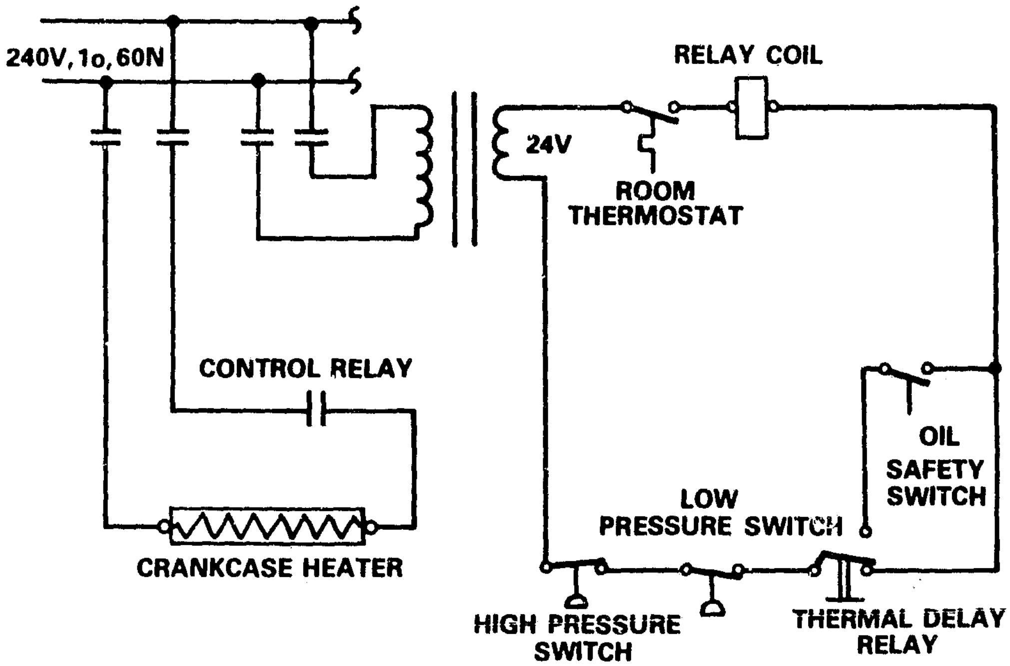

Figure 4-41. Residential Refrigeration Module Control Circuitry

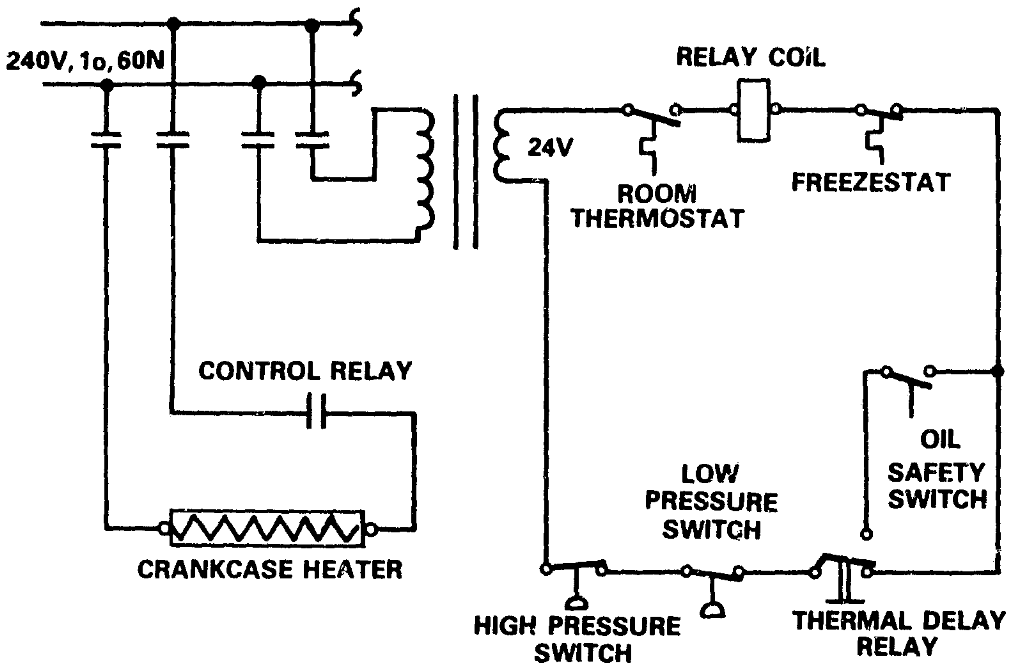

Figure 4-42. Multiple-Family/Commercial Refrigeration Module Control Circuitry

- Multiple-family/Commercial Subsystem:
  - Capacity - 300,000 Btuh
  - Bhp Input - 19.6 Bhp
  - COP - 6.0**

Sound rating numbers will not be a set objective under this project. There will be no sound numbers established by testing; nevertheless, Lennox will remain sound conscious during all phases of equipment design. To give example of sound conscious design, three key factors will be incorporated into the units: 1) The units will be completely encased by cabinetry; 2) Rotating mechanical components will be isolated from the main body of the unit by vibration dampeners; and 3) Fiberglass insulation will line the entire cabinet.

4.12.3 Rankine-Cycle Subsystem

The Rankine-cycle subsystem power loop is shown in Figure 4-43. This section discusses the working fluid selection study and the preliminary hardware investigation that has been conducted.

4.12.3.1 Working Fluid Considerations -- Two fluids were considered to be viable candidates for use in the Rankine cycle: Refrigerants 113 and 11 (R-113 and R-11). Both are relatively high-molecular-weight fluorinated and chlorinated carbon compounds. Many criteria were used to narrow the large fluid candidate field to these two. The most significant considerations were cycle efficiency, thermodynamic and transport properties, turbomachinery considerations, toxicity, and flammability.

** Per ARI Standard 590-76

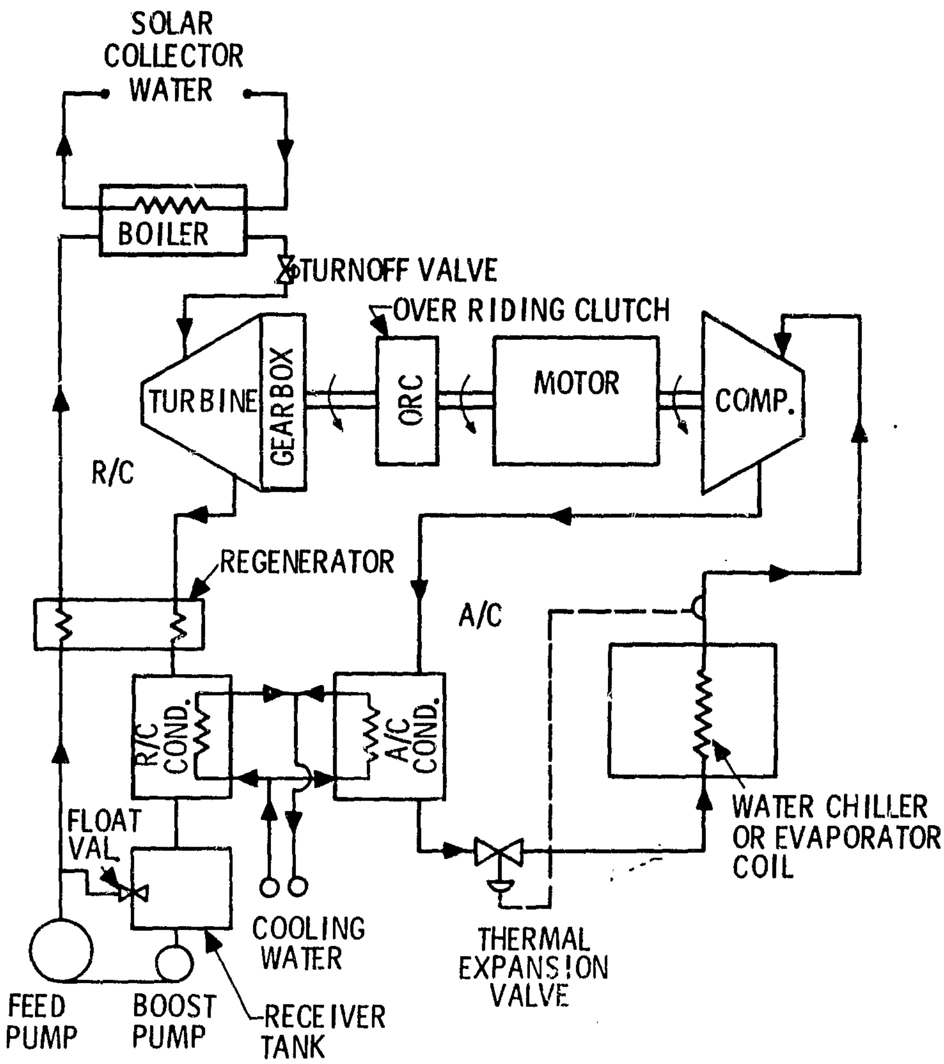

Figure 4-43. General System Schematic for Rankine-Cycle Air-Conditioning System

Comparing the two fluids in the temperature range shown in Table 4-4, a 5 percent advantage in cycle efficiency is obtained with R-113. To attain this performance, however, the R-113 requires a regenerator to recover some of the superheat remaining in the turbine exhaust vapor. The R-11 cycle, due to the fluid's thermodynamic properties, contains little or no superheat in the turbine exhaust and so does not require a regenerator.

For the same operating temperatures, R-11 operates at a slightly higher pressure than R-113, as shown in Table 4-4.

Table 4-4. Fluid Operating Temperature Comparison

Heat Exchanger	Fluid	
	R-113	R-11
Boiler pressure at 190°F	47 psia	90 psia
Condenser pressure at 95°F	10 psia	22 psia

This fact favors R-11 because low-pressure condenser systems generally are less attractive from a leakage standpoint.

The same cycle operating conditions also produce different turbomachinery for R-113 and R-11 systems. For the 190°F boiling and 95°F condensing cycle, an R-113 turbine demonstrating 72 percent efficiency and producing 3 horsepower rotates at 39,000 rpm and is 2.4 inches in pitch diameter, while an R-11 turbine with the same power and efficiency is a 60,000-rpm, 1.6-inch unit. Since gearbox losses increase rapidly with speed, a slower R-113 turbine with the same efficiency is much more attractive. For the 3-ton system, the best R/C fluid choice is unquestionably R-113. Its higher cycle efficiency and turbomachinery advantages clearly offset the disadvantages of the regenerator and the below atmosphere condenser.

In the 25-horsepower range, a 77 percent efficient R-113 turbine has a 22,000-rpm, 5.1-inch rotor, while an equally efficient R-11 turbine turns at 33,400 rpm and is 3.6 inches in diameter. However, a R-11 turbine rotating at the same speed as the 77 percent R-113 machine (22,000 rpm) would sacrifice only two points in efficiency with the speed reduction, and the R-11 system would not require a regenerator. For these reasons, the best fluid system choice for the multiple-family dwelling (25-ton air conditioner) is unclear. It is concluded that both R-113 and R-11 would offer reasonable approaches. R-113 was selected from the simulation study for the 25-ton development system, however, due to its slightly higher overall performance.

4.12.3.2 Three-Ton Hardware -- The following sections describe in some detail the status of the selection process for various pieces of hardware for the 3-ton unit.

4.12.3.2.1 Turbine -- A usual approach to achieve optimum system efficiencies in a single-fluid hermetic system would be to use an expander of the same type as the compressor (i.e., a piston expander with a piston compressor and a turbine expander with a centrifugal compressor). For the present 3-ton system, however, a turbine prime mover was selected because of the lower cost and higher prototype reliability when compared with a positive-displacement expander. Barber-Nichols has had extensive experience in the design and fabrication of turbomachinery for use in systems similar to the present application. The measured performance of a 2-inch turbine was reported by Robert Barber.* A design point efficiency of 72 percent was achieved by this small radial inflow turbine.

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* Barber, Robert E., "Solar Air-Conditioning Systems Using Rankine-Power Cycles - Design and Test Results of Prototype 3-Ton Unit," Barber-Nichols Engr. Co., Arvada, Colorado.

A preliminary turbine performance analysis was completed using the non-dimensional parameters of specific speed (N_s) and specific diameter (D_s). These parameters relate system cycle conditions (flowrate and available energy) and design variables (rotational speed and turbine diameter) to achievable performance for various turbine concepts. To perform this preliminary analysis, a turbine speed of 35,000 rpm (based on a tradeoff study relating turbine efficiency, size, and gearbox loss to turbine rotational speed) was selected for the 3-ton unit. The above similarity analysis and tradeoff study resulted in a 2.5-inch turbine.

The turbine type offering the best potential performance for selected design conditions is a radial inflow turbine. This type of turbine with the selected design conditions was found to match well with previous turbines designed and developed by Barber-Nichols. The turbine rotor is similar to that found in automobile turbochargers. In this application, existing turbine rotor and housing hardware can be modified to result in an extremely cost-effective turbine prime mover capable of efficiencies greater than 70 percent.

A preliminary estimate of off-design turbine performance was then made for a range of Rankine-cycle off-design conditions. The primary measure of turbine efficiency is the ratio of turbine wheel tip speed to the working fluid isentropic spouting velocity (U/C_o), where:

$U = N\pi D/720$

N = turbine rotational speed, rpm

D = wheel tip diameter, in.

$C_o = \sqrt{2gJ\Delta H'}$

and

$\Delta H'$ = overall isentropic head, Btu/lb

As may be noted, the ratio U/C_o accounts for changes in both turbine rotational speed and cycle off-design parameters affecting the overall isentropic head. Figure 4-44 presents the variation in turbine efficiency with off-design U/C_o. The figure was constructed using data obtained previously with a radial inflow turbine similar to that proposed for the present application. This procedure is intended to account for first-order influences on turbine efficiency. Secondary changes due to off-design pressure ratio, Reynolds number, and Mach number will be evaluated during the final turbine performance design analysis.

Final turbine design parameters, including nozzle and rotor definition, will be completed during the final design. Barber-Nichols will rely heavily on tested performance of these components in selecting an optimum turbine configuration. Design and off-design turbine performance will also be predicted during this phase using a digital computer program written for that purpose.

4.12.3.2.2 Gearbox -- The following specifications have been outlined as preliminary requirements for the 3-ton-unit gearbox. The input speed to the gearbox will be approximately 35,000 rpm, the output shaft speed 1750 rpm or lower depending on compressor selection, and the maximum gearbox power loss of 0.2 horsepower. The gearbox will have a hermetic seal either at the high-speed shaft or the low-speed shaft to contain the working fluid. The life expectancy should be in excess of 3000 hours. The gearbox should be designed to allow minimum production costs. Its power throughput will be approximately 3 horsepower.

Several approaches have been considered for meeting the above specification requirements. These approaches consider the development time available and the probable risk in reaching the above design goals. Previous designs have shed some light as to possible paths that can be taken.

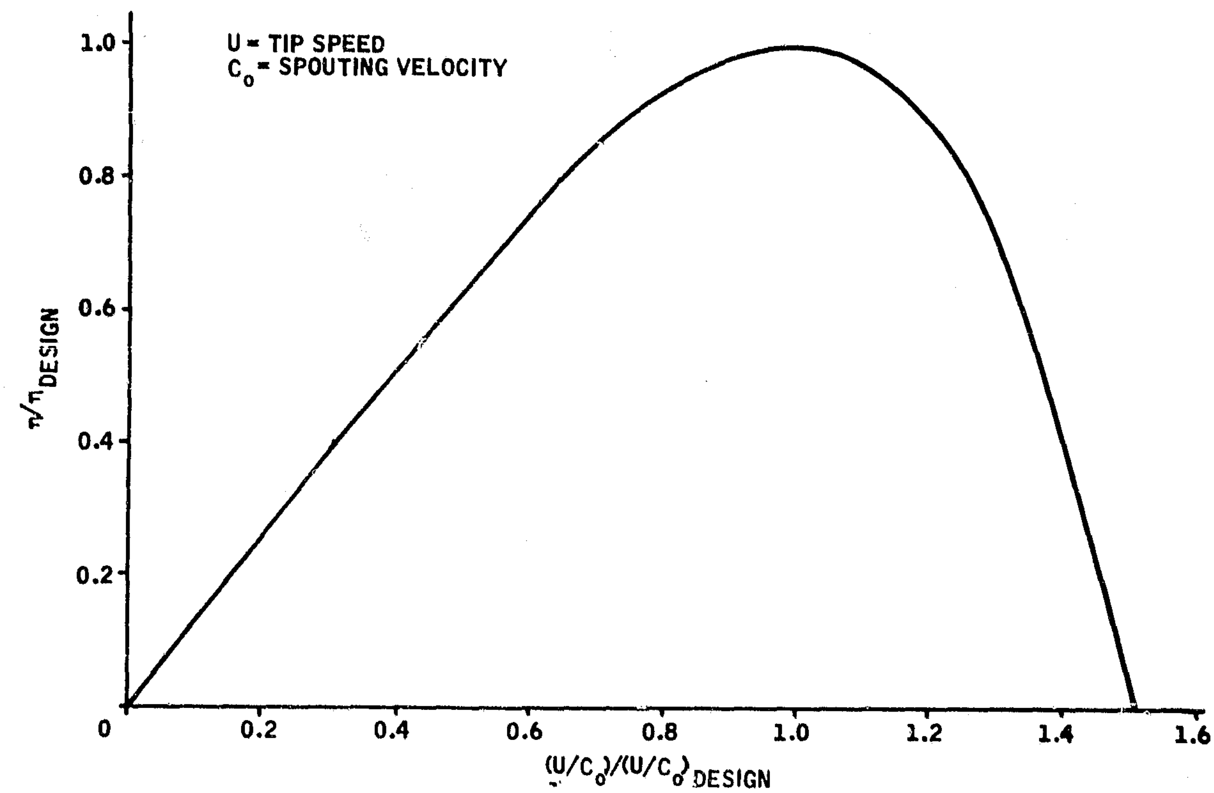

Figure 4-44. Variation in Turbine Efficiency with Off-Design U/C_o

Two previous 3-ton air-conditioning units have been in operation for various periods of time. Each of these units used a commercially available gearbox modified for high-speed service. The first unit developed had a double-face seal on the output shaft, thus providing the hermetic seal at that point. It was also canned inside a liquid-filled chamber that acted as a heat sink for cooling. The high-speed turbine shaft had a single-face seal to prevent gross working fluid migration through the turbine end of the gearbox. The gearbox was vented to the low-pressure condenser to prevent liquid buildup in the gears. In the design of the second unit the hermetic seal was moved to the high-speed shaft assembly to allow the gearbox to be vented to ambient air and, thus, it only required a low-pressure lip seal on the output shaft. Both of these units used a splash lube system developed to minimize power loss. The second unit did not have a cooling chamber around it but used a cold-water plate bolted to the bottom of the gearbox.

The primary advantage in using the commercially available gearboxes was in the availability of existing precision gearing that could operate at high speeds with minimum modification. These units were rugged and oversized, resulting in minimum development time. The advantage of the first unit over the second lies in the area of cooling. There was more than sufficient cooling available to handle the power losses of that gearbox. The reason this unit was not used in the second system was because of the power loss due to the high-density working fluid vapor in the gearbox chamber. Three face seals in the assembly also contributed to this higher loss. Initially there was a chance that oil migration from the gearbox would never return to the gearbox but would remain in the rest of the system. It was found during the operation of the second unit that it was much noisier than the first unit. This occurred because of considerable noise damping available in the liquid chamber surrounding the gearbox in the first unit. Also, the cold-plate-mounted gearbox was just marginally cooled, and any attempts at adding sound damping around the unit resulted in overtemperature.

Another approach is under consideration in an effort to solve the above problems. It was noted that there are many high-reduction gearboxes on the market in the hand and power tool industry that are presently in production and from which production costs could be extracted on a high-volume basis. One such unit that appears to be very well adapted to this application is a unit made by Milwaukee Tool Company that has a high-speed motor driving a diamond drill unit for cutting holes in concrete. This unit is designed as a commercial product and its rated capacity in that particular product is 2 horsepower with 1000-rpm output. Its input speed is 20,000 rpm, and it is a triple-reduction unit. This unit has smaller gears than the other units and because of this has a lower power loss. Projected power losses for this unit are: 0.2 horsepower, including seals, thus meeting specifications. Gear tangential velocities are very low in this unit, thus presenting no windage loss problems if a low-speed hermetic seal is used. It has the build-in capability of changing from one speed to another which may be modified to allow an internal clutching mechanism that would disengage the turbine during motor operation. The unit is presently in production in quantities of 500 or more per run, and existing component costs are comparable with previously designed units. The advantage of using this unit in this particular system lies primarily in the low power loss. Additionally, production costs can be very closely estimated. The primary disadvantage of using this unit in the proposed system lies in the fact that the unit has not been previously used in a similar application.

There are several possible techniques for lubricating and cooling the gearbox. One is a conventional approach of using a shaft-driven gear pump to pump fluid from a sump through a cooler-filter and on through various lube jets directed at the bearings and gears. The other system developed uses the gearbox with a splash system in which the oil is collected in the sump of the gearbox and splashed over various components in the gearbox for lubrication and cooling. This will require a method of cooling the oil. This latter system is the one used in the previous 3-ton air-conditioner units.

Another approach that will be investigated is a mist lubrication/cooling system. This would greatly reduce the losses in the bearings. The primary problems in such systems are generating the mist and providing some kind of throughflow to provide the cooling. The system under study for this program uses a gas-driven mist generator similar to those used in paint spray systems. The driving gas (hot Freon) is bled off at the boiler. This gas is metered through a control valve and is then expanded through a standard paint-spraying nozzle. This spraying nozzle provides a siphoning effect, sucking up oil and generating a mist which is sprayed into the gearbox. The gearbox sump, which is external to the gears themselves, is vented to the condenser, allowing the gas introduced with the spray to migrate to the condenser. The gearbox is cooled by direct contact with liquid Freon from the boost pump being injected directly into the oil.

The system does require two primary control valves. One valve controls the proper gas flow through the mist-generating nozzle by maintaining a certain pressure drop across the mist-generating nozzle. The cooling is controlled by the second valve which is a thermal valve that cycles on and off to maintain a certain temperature in the oil reservoir. A primary advantage in such a system lies in the fact that all components involved in mist-generating and cooling of the gearbox are static systems that are not subject to reliability problems and premature failure.

One development item in this particular system is the control of the oil migration back to the gearbox. Air-conditioning systems currently operate under similar conditions, and it is proposed that a conventional approach be followed.

4.12.3.2.3 Boiler Feed Pump -- There are several possible approaches for the boiler feed pump. The primary consideration is whether to use a direct-driven or auxiliary-motor-driven unit. A second choice would be positive-displacement or centrifugal. The means of driving affects the start-up characteristics of the cycle (requirement for start-up pump) and the

possible efficiency (speed) of the pump. The second choice is related to control methods and matching characteristics of the components for off-design operation. Barber-Nichols has experience with all types of boiler feed pumps.

A boiler feed pump candidate that was recently acquired is a diaphragm pump. These have been used with Freon but not R-113 specifically. They are very-low-speed units that must be driven by a belt reduction system or a gear reducer. In discussions with the distributor's engineers, the shortest life components are the Viton diaphragms which are presently rated at 1500 hours at the pump's maximum rated speed and pressure. Since lower speeds and lower pressures can greatly increase life, it was recommended that the pump be tested for our application operating at a lower speed. Since the pump will be operating at a lower speed and at lower pressure, the life values should improve dramatically. This pump will be tested to verify performance and NPSH requirements. Then, possibly, some simple modifications to the pump will be apparent that will improve characteristics and possibly also add additional life.

A boost pump is applied in the system to provide a sufficient net positive suction head (NPSH) to meet the requirements of the main boiler feed pump. The pump is located at the discharge of the condenser in the system hot well. A bypass line is provided around the pumps to assure a quantity of liquid to the pump and provide stability in system operation.

Two off-the-shelf candidate boost pumps are currently under test at the Barber-Nichols facility. Manufacturers of these pumps are Sundstrand and Grundfos. Both units are hermetic, and both use a wet motor. The bearings are lubricated by the working fluid. These inexpensive pumps were designed for use in hot-water systems, and no mechanical modifications are anticipated to use either pump in R-113.

The pump impellers are designed with backward-curved vanes. A typical characteristic curve, with the delivered head increasing at low pump flow-rates, was found for both pumps during testing. This characteristic will provide quite stable pump operation over the range of flowrate.

4.12.3.2.4 Motor -- The 3-ton air-conditioning system will use a nominal 3-horsepower, single-phase motor. The typical efficiency for a production motor in this category is around 73 percent. These motors will retain this efficiency until loads below 50 percent are reached, at which time the efficiency will decrease steadily until the "no load" condition is attained. Typical "no load" power consumption will be around 0.3 kW. These motor efficiencies do contribute to the overall system efficiency and any improvement also improves the overall solar energy conversion efficiency. Single-phase motors of this power rating with efficiencies as high as 90 percent can be designed and fabricated. A high-efficiency motor such as this would be expensive, and at this time, Barber-Nichols is not aware that any of the manufacturers of production-quantity motors are considering the design or marketing of a high-efficiency single-phase motor.

4.12.3.2.5 Controls -- The basic Rankine-cycle power system requires only one active component for complete control. These power systems tend to be self-balancing and can be adjusted to run without any active control components. To handle a wide range of off-design operating conditions efficiently, a float-activated bypass valve is used to match pump delivery to system flow requirements. The pump characteristics at off-design conditions are the only factors that require adjustment to keep the system balanced or operating at its best performance. As shown on the system schematic, Figure 4-43, a reservoir collects the condensed fluid and supplies it to the pumps for the system. A float bypass valve is placed in this tank and tries to hold the reservoir level fixed by increasing or decreasing the bypass flow from the pump back to the reservoir. This action tends to balance the total system fluid inventory and thus aids the self-regulation of the system. This bypass

valve performs a second function during high-power-level operation and keeps the pump from depleting the reservoir and cavitating the pump. The bypass also allows the pump to be somewhat oversized and able to provide the increased flows when required. This reservoir-level-control approach, coupled with the proper fluid charge, enables a system to operate from a basically zero power level up to a maximum limited by the capacity of the pump to meet required head/flow requirements.

In addition to the basic Rankine-cycle loop control, a system requires a control logic center to integrate the Rankine-cycle system and the air conditioner with the rest of the solar-energy collection system and the building temperature control. This logic center will cycle the air conditioner on and off with or without the Rankine-cycle power system to meet the commands received, while monitoring various performance parameters to protect the combined cooling system from damage in the event of equipment malfunctions. The logic control for these Rankine-cycle-powered cooling systems will be composed of simple relay logic using switch-type sensors for the required inputs and for sending electric power to the various control components required.

Each cooling system will be controlled primarily by an on or off command from a central solar controller. A voltage will be supplied from this controller whenever the building requires cooling. Upon receipt of this command, the logic will activate the air conditioner by turning on either the electric motor or the Rankine-cycle power system or both. The decision to use the Rankine-cycle power will be a function of the availability of solar heating water above a minimum-temperature level. The performance studies being made at the present time will establish the requirement or basis for establishing the various combinations of power source use desired. If the studies indicate that supplementary electrical power is needed over a significant portion of the cooling day, then there will probable be no benefit in providing a control mode for using the Rankine-cycle power system with the electric-motor shutoff. If the study shows that there is a significant performance

advantage to running with the Rankine-cycle power system only, then logic will be required to select this mode of operation. One of the simplest approaches to provide logic for the decision choosing Rankine-cycle power only or both power sources is to use a two-level thermostat for the building. With a two-level thermostat, two separate switch closures are provided with a finite temperature difference between them. The first switch closure will be used to command the Rankine-cycle power system to power the air conditioner. If the Rankine-cycle power is sufficient to support the cooling load, then the thermostat will work around this setpoint. If the cooling load is too large for the Rankine-cycle power system, building temperature will climb until it activates the second switch which, in turn, will energize the electric motor and it will pick up its share of the air-conditioning load. Building temperature then will cycle about the second setpoint. The decision to use the Rankine-cycle power upon a command for cooling is dependent only on the availability of solar water hot enough to provide useful power for the system. The energizing of the Rankine-cycle system for cooling would thus be overridden by a temperature switch that would allow the Rankine-cycle system to come on if the water temperature is high enough and shut it off if too low. This logic panel, at a minimum, would control the motor starter and the solenoid valve between the Rankine-cycle boiler and the turbine. If, in the development of the final system configuration, a clutch is used to isolate the electrical motor when the Rankine-cycle system is on, the control center will control this clutch in parallel with the Rankine-cycle solenoid valve.

Both the Rankine-cycle power system and the air-conditioning system have physical parameters that require monitoring for overall system protection. In the air-conditioning system, the typical parameters monitored to ensure proper performance are compressor discharge pressure, suction pressure, and compressor oil pressure. The Rankine-cycle system parameters monitored will be lube oil pressure and lube oil temperature. Sensors will be provided to monitor these parameters and will be used as interlocks in the logic chain to prevent system start-up or provide system shutdown in the event their setpoints are exceeded.

4.12.3.2.6 Parasitic Losses -- For the Rankine-cycle system to operate, support is required from various electrically-powered accessories. The 3-ton demonstration unit built for the Honeywell van uses an electric clutch, solenoid valve, boost pump, and a logic control box with relays and motor starter. On the demonstration model, these accessories use a total of 170 watts of power. The boost pump requires 70 watts, the clutch 40 watts, and the relay logic control box 50 watts. Fifty watts is not a true reflection of the control logic requirements because at least 35 of these watts are consumed in a DC power supply required to power the clutch. The control approach, and thus the components required, has not as yet been established for this system. The electrical clutch used in the demonstration system was included so the Rankine-cycle power system could drive the induction motor in a generating mode. This generating mode is not being considered for these 3-ton systems. The elimination of the clutch can save 75 watts of this parasitic power. There is a possibility, however, that an electrically-operated clutch will provide overall performance improvement if used to decouple the turbo gearbox when there is insufficient power from the Rankine cycle power system to drive the air conditioner compressor. This clutch would not have to be electrically-powered but could be mechanical with solenoid operation. The control approach used in the overall system configuration will be based on minimizing these parasitic power requirements while achieving overall system performance objectives.

There are other power system parasitic losses besides the electric ones that have been mentioned. The gearboxes required to reduce or match the turbine to the electric-motor speed will consume approximately 0.2 horsepower with the 3-ton unit. If, in the design of the cooling system compressors, the selection requires a nonstandard motor speed, a belt-drive reduction system will be required. Power losses in these belt drives are typically 2 percent or less of the power level transmitted. A physical packaging configuration that requires jack shafts or idlers to control the belts will add even more loss to the mechanical power transmission system. In the A/C operating

mode, where the Rankine-cycle power system is not producing power and all of the power is supplied by the electric motor, there is a need to decouple the Rankine-cycle drive from the motor and compressor. There is a power loss associated with driving the turbine backwards through the gearbox. One technique is to use an overrunning clutch that prevents power feedback into the gearbox. These overrunning clutches, however, also have parasitic losses when used in the override mode. All of these loss factors tend to reduce the overall efficiency of this Rankine-cycle-powered air-conditioning system. Each one of these loss factors will be evaluated carefully and design approaches selected that minimize their effect on the performance of the cooling subsystem.

4.12.3.2.7 Rankine-Cycle Heat Exchangers -- Barber-Nichols' time-proven approach for selecting low-temperature Rankine-cycle heat exchangers is to use commercially available units wherever possible, thereby reducing cost, lead time, and development. The preheater, boiler, and condenser will be purchased from nationally recognized vendors. We are not aware of any commercially available heat exchangers suitable for the Rankine-cycle regenerator. This unit therefore will be designed and assembled by Barber-Nichols.

Prices for these types of heat exchangers were used to estimate heat-exchanger costs during the R/C optimization discussed in Subsection 6.11.5. Additional features of the heat exchangers are discussed below:

- Preheater -- The preheater is a commercially available industrial-type heat exchanger commonly used as an oil cooler, with water as the heat sink. For this application, solar collector water is placed in the tubes and the R/C working fluid flows with low-pressure drop through the shell. A one-pass, counter-flow configuration is used to maximize heat transfer with minimum cost.

The tube-side heat transfer surface may be mechanically cleaned without disturbing the shell-side piping. The fixed tube sheets are welded or brazed to the shell (depending on manufacturer) which eliminates any joints on the R/C side of the unit. These units are designed by their manufacturer for temperatures and pressures in excess of our requirements and their large production quantities result in low cost. Barber-Nichols' design approach has been verified by testing to improve confidence in our predicted performance.

- Boiler -- The boiler transfers heat from water to a boiling organic fluid. This heat exchanger is functionally very similar to water chillers commonly found in air-conditioning systems. In fact, Barber-Nichols' recommended approach is to use a commercially available water chiller for the R/C boiler. This recommendation is based on our experience with low-temperature Rankine cycles using this type boiler. Forced-convection designs are preferred because the relatively small water-to-organic fluid temperature differences anticipated for this application are too small to produce the natural convention required in pool-type boilers. We have had excellent experience with the forced-convection evaporators.

These units are designed for use with R-12 or R-22 and, therefore, have a pressure capability well in excess of that required with R-113. This will enhance system reliability and safety. The large production quantities of these readily available heat exchangers result in low cost and short lead times. A variety of types and configurations are available to suit our design requirements. These units are designed to have low-pressure drops and are compatible with the various Freons and with water and water-glycol solutions.

C-2

- Condenser -- The R/C condensers have requirements very similar to condensers for air conditioning and refrigeration units. They both desuperheat and condense a fluorocarbon fluid using water as the heat sink. Therefore, commercially available air conditioning and refrigeration condensers are proposed for the Rankine-cycle condensers. These units are designed for 300 psig or more and their high pressure capability will greatly enhance system reliability and safety. Their welded tube sheets eliminate gasketed joints on the Freon side, reducing leaks and simplifying maintenance. Several designs feature all copper water channels and an epoxy-coated tube sheet and water plate to prevent pitting caused by galvanic action. Units feature mechanically cleanable tubes and low pressure drops. Units are designed for use with cooling towers.

- Regenerator -- The regenerator is a vapor-to-liquid heat exchanger with a high effectiveness. Because of the required low-pressure drops and large required heat-transfer area, we are unaware of suitable commercially available units. Therefore, Barber-Nichols' approach is to design and build a heat exchanger specifically for this application.

  This heat exchanger uses a heavy-duty industrial oil cooler core that is specially manufactured to Barber-Nichols' specifications. The design incorporates copper tubes for high heat transfer and economical assembly, and aluminum fins because of their light weight, low cost, and excellent heat-transfer characteristics. The external fins incorporate serrations for strength and durability, and incorporate a sunburst collar to impart turbulence and increase heat transfer.

Barber-Nichols has built several of these units and has test data to back up our extensive analysis to ensure the best heat-exchanger design possible. Test data and analysis show that this concept results in high heat transfer and extremely low pressure drops.

The heat-exchanger shell is designed in accordance with Section VII, Division I, of the ASME Pressure Vessel Code. The units will withstand full vacuum on both the tube and shell sides. This capability enhances reliability by simplifying system controls and valve requirements. Each unit is fully pressure-tested before it is connected to the system. The regenerators incorporate fittings designed for the particular application to minimize joints and fittings.

Similar units have been built and used in numerous systems and their superior performance and reliability have been verified.

4.12.3.3 Twenty-five Ton Hardware -- The following describes in some detail the status of the selection process for various pieces of hardware for the 25-ton unit. In most cases this is basically the same as for the 3-ton and will not be repeated here. The areas that are different will be discussed.

4.12.3.3.1 Turbine -- The turbine for the 25-ton unit will be similar to that for the 3-ton unit, differing in speed and diameter. There will be a slight improvement in efficiency to about 77 percent.

4.12.3.3.2 Gearbox -- The gearbox under consideration for the 25-ton Rankine-cycle air-conditioning system is an existing-design gearbox used in other Rankine-cycle turbine-driven units under previous development by Barber-Nichols. The unit is basically a ground-gear, double-reduction

gearbox using a wet sump with a positive-pressure oil system. This unit is oversized for the 25-ton unit and, as such, has proportionally large losses (2 horsepower). These losses can be reduced through further development. One area that could prove fruitful is to use a mist lube system as mentioned in the previous 3-ton air-conditioning section. Another area would be to reduce seal loss by using a low-speed seal rather than a high-speed seal.

Depending on the results of development tests with the 3-ton gearbox, a gearbox unit is available from Milwaukee Tool Company that may be adaptable to this power range. This unit could be modified as previously stated for the 3-ton unit, thus providing a low-loss unit.

4.12.3.3.3 Pumps -- The main boiler feed pump for the 25-ton unit will probably be driven by the gearbox. An efficiency of 60 percent is expected for this pump. The boost pump and start-up pump, if required, will be similar to those for the 3-ton unit.

4.12.3.3.4 Motor -- The 25-ton air-conditioning system requires a 20-horsepower, three-phase motor. Typical efficiencies for this category of motor are around 86 percent at full load, with this efficiency being maintained to power levels of 50 percent. The Century Electric Division of Gould has produced and is presently marketing a new, high-efficiency, three-phase motor. A 20-horsepower motor from the Gould "E plus" line has a guaranteed efficiency of 90.5 percent at full load. These motors retain their efficiency down to the 50 percent load also. A secondary benefit achieved with this motor line is an improved power factor. Power factors of around 0.9 are achieved as compared with 0.8 for a standard production unit today. This improvement in performance results in a cost factor of 1.25 for these motors over the standard production model.

4.12.3.3.5 Controls -- Same as for 3-ton system.

4.12.3.3.6 Parasitic Losses -- With a 25-ton air-conditioning system, the same accessories as for the 3-ton system are required. The boost pump naturally is bigger and will require up to 500 watts of power. Controls power should be identical to that required in the 3-ton system. A clutch, if required for this larger system, will have to be of a mechanical configuration and solenoid-actuated at these higher power levels. With the proper clutch configuration, the solenoid-actuator will require power only when engaging and disengaging the clutch. The gearbox loss for the 25-ton unit will be under 2 horsepower.

The other losses described in the section on parasitic losses in the 3-ton unit also pertain to the 25-ton unit and will not be repeated here.

4.12.3.3.7 Heat Exchangers -- Same as for the 3-ton system.

4.12.4 Rankine-Cycle Optimization Study

The optimization study by Honeywell with inputs from Lennox and Barber-Nichols involved carrying out cycle computations to determine power output and efficiency at various operating conditions. System characteristics provided the basic data necessary for the evaluation of an optimum design. A number of parameters were considered in the selection and development of the baseline schematic including:

- Cycle temperatures that are compatible with present day flat-plate technology. This requirement limited the inlet temperature to the engine boiler to a maximum of about 220°F. Various design temperatures below this upper limit were investigated.

- Engine design and size with the minimum auxiliary power requirement over the range of operating conditions. The engines were coupled to small-to-medium-capacity air conditioners. Design capacities of 3 tons and 25 tons were selected for the single-family and multiple-family residences, respectively.

- Rankine-engine working fluid based on the temperature limitations of the cycle and its thermodynamic states. The refrigerant contribution to the requirement of minimal maintenance on the unit over a long operating life was also considered. Both R-113 and R-11 were studied for the medium-size engines.

- System control to optimize the use of collected solar energy and minimize auxiliary power requirement. Control reliability and simplicity in interfacing with the overall system were also factors in the evaluation criteria. In the present study, two control schemes were investigated: constant-speed and

variable-speed. Basically, in constant-speed operation, auxiliary power from the electric motor is supplied to maintain the air conditioner at its design capacity. In the variable-speed mode, the cooling capacity of the unit varies with the fluid inlet temperature and is electrically enhanced only when it fails to meet the space cooling requirements.

The effects of these parameters and various design values were studied through component and system models incorporated in a year-long simulation. Additional cycle computations were then made based on the operating condition selected from the simulation. These last cycle conditions then established some cost figures for the cycle. The subsections that follow describe these computations in some detail and present a summary of the major results of the simulation.

4.12.4.1 Off-Design Study -- The cycle off-design operation was analyzed to determine the performance during the time that insufficient insolation was available to operate at the design conditions. These data were the input to the simulation program which determined the auxiliary power required to operate the air conditioner at design capacity. This off-design study was completed for a range of design collector-exit temperatures from 160°F to 210°F. At each collector-exit design temperature the off-design operation performance was determined for a collector-exit temperature of 10°F above the design value and as much as 40°F below the design value.

The off-design study involved a complete cycle calculation at the design condition (collector-exit temperature), and then the cycle was recomputed using the following described assumptions as to the off-design operation of each component. The cycle calculation for the design study will be described first, then the assumption for the off-design computation will be discussed.

The basic cycle was assumed to operate between the temperature limits of the collector-exit design temperature and the temperature of the cooling tower water at 85°F. The design flowrates of the hot source and cold sink were 4 gpm/ton and 3 gpm/15,000 Btu/hr, respectively. A 10°F approach temperature was assumed in both the boiler and the condenser. The pressure loss on the working side was assumed to be 5 percent of the absolute pressure on both the high-pressure side and the low-pressure side of the cycle. The working fluid was R-113 for the 3-ton unit. This choice of working fluid implied the use of a regenerator which had an assumed effectiveness of 0.80. The efficiency of the turbine and pumps were assumed to be 0.72 and 0.50, respectively. This is based on past experience and the analysis described in the turbine and pump sections of this report. The gearbox loss (0.2 horsepower) was included in the analysis, so that the design power is the power out of the gearbox. The design power was 1, 2, 3 and 4 hp for the 3-ton unit. The results of this design-cycle computation provided the baseline conditions for the off-design performance calculation.

The typical off-design operating conditions result from the lack of sufficient solar input which the cycle experiences at a lower than design collector-exit temperature. A similar off-design condition is possible during periods of high solar insolation but with low cooling requirement. The collector-exit temperature would then be above the design condition. In either case, the change in the high temperature of the cycle, with the assumption of hardware fixed by the design condition, will change the performance of the cycle.

Basing the computation on the temperature of the working fluid as it leaves the boiler (rather than the collector-exit temperature), the working fluid temperature was allowed to increase or decrease by some amount, such as 10°F, to represent the off-design condition. This then defined the boiler pressure (saturated) and the pressure input to the turbine. The flow through the turbine was determined by the choked condition in the nozzles. Therefore, the change of the flow due to the change in the cycle conditions could be

determined. The boiler-and-condenser-approach temperature difference is assumed to vary directly with the flowrate. This, in turn, defined the collector temperature and condenser coolant temperature. The frictional pressure drop in the heat exchangers varied with the square of the flowrate. The turbine efficiency varied with the isentropic spouting velocity as described in the turbine section. The pump efficiency was assumed to remain constant but the pump work varied with the head rise. The actual flow in the pump was the design flow, since the speed was constant. Any excess was bled back around the pump.

The off-design computations were carried out for the same conditions as described for the design condition. These are presented in Table 4-5. The plot for the power output and the Rankine-cycle efficiency is shown in Figures 4-45 and 4-46 for a design collector-exit temperature of 190°F. This represents the typical plot, and the curves for all the other conditions are similar in shape. These curves were used as the input data for the Honeywell simulation program to determine the optimum design temperature and power level. Added to these graphs are curves for a power level of 2.3575 horsepower which was the power selected by the simulation study.

Table 4-5. Off-Design Study Power Levels
(Values in Horsepower)

Cycle	Design-Point Collector Exit Temperature, °F					
	170	180	190	200	210	220
3-Ton/R-113	1, 2, 3, 4	1, 2, 3, 4	1, 2, 3, 4	1, 2, 3, 4	1, 2, 3, 4	1, 2, 3, 4
25-Ton/R-113	15, 25, 25	15, 20, 25	15, 20, 25	15, 20, 25	15, 20, 25	15, 20, 25
25-Ton/R-11	15, 20, 25	15, 20, 25	15, 20, 25	15, 20, 25	15, 20, 25	15, 20, 25

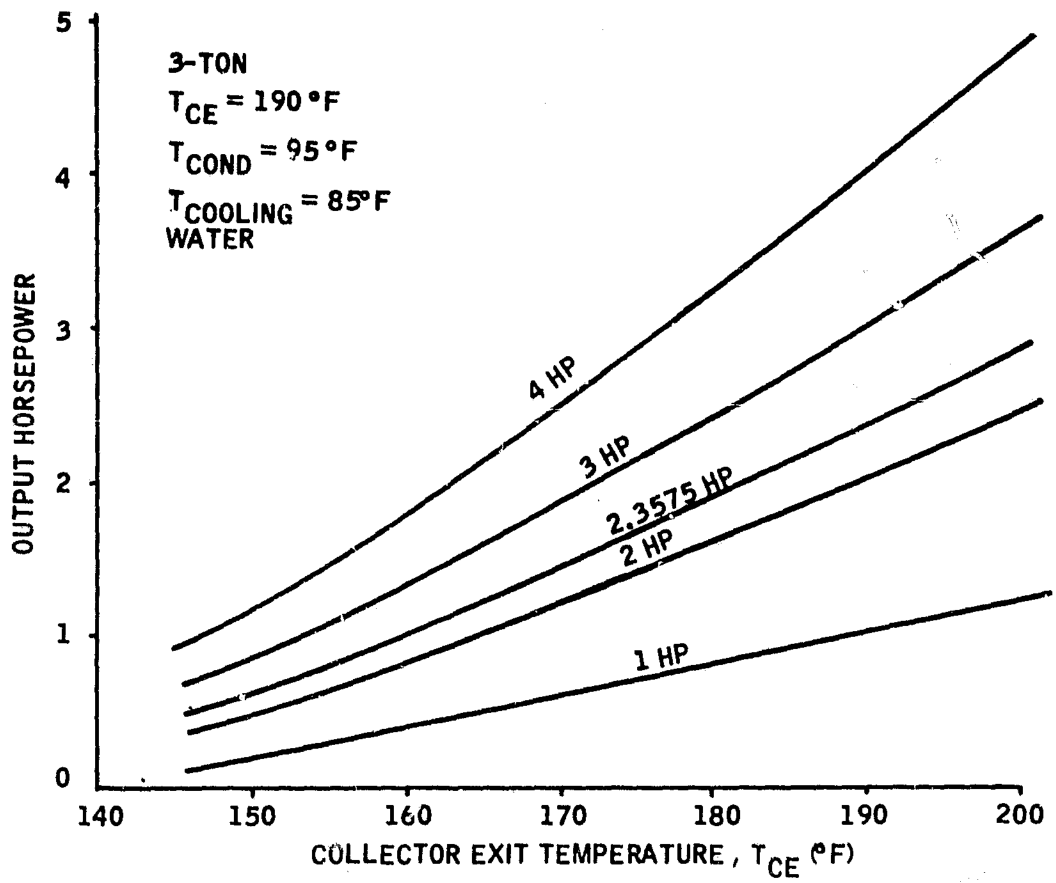

Figure 4-45. Output Horsepower versus T_{CE} at Constant-Horsepower Off-Design

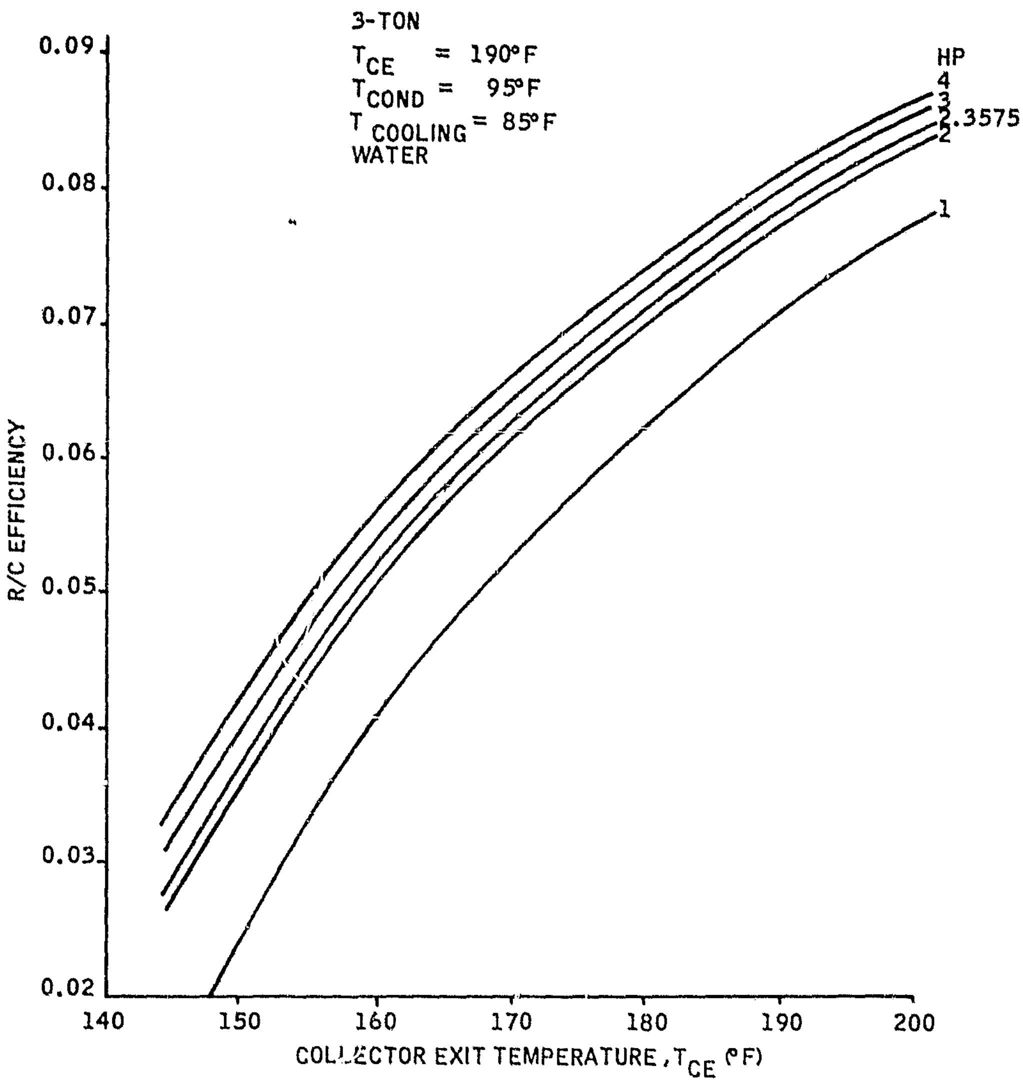

Figure 4-46. Rankine-Cycle Efficiency versus T_{CE} at Constant-Horsepower Off-Design

4.12.4.2 Simulation Study and Results -- The results of the off-design study were input into the Honeywell simulation program to determine an optimum collector temperature and optimum power level. Along with the power output and efficiency, the feed pump capacity and the control methodology are two other aspects of Rankine-cycle operation that can affect the results. These were also input into the program.

One significant limiting facet of the Rankine-cycle operation is the capacity of the boiler feed pump. Usually, this limit is 10 percent over design but it can be made any desired value. It is a question of trading off design performance with maximum capacity. The effect this has on the simulation is that the Rankine-cycle operation (power output and efficiency) levels fall off at some point above the design point rather than just continuing on up the curves presented. This was input at 10°F above the design point.

The other major factor input to the program was the method of controlling the cycle. One such method is to use the motor as a generator to limit the speed when an excess of solar energy is available. Along with this there are a number of alternatives, such as variable-speed operation when the off-design condition allows the Rankine-cycle air conditioner to move to off-design speed. Then, at some selected point, the motor/generator would cut in to bring the speed back to the design level.

Figure 4-47 shows the effects of various engine sizes and design temperatures on the yearly consumption of auxiliary power needed to meet the cooling requirements of a single-family residence in Atlanta. With a compressor COP of 7.0 the 2-horsepower engine provides the design power requirement of a 3-ton air conditioner. The results of the simulation for a fixed collector area of 756 ft^2 show that the heat addition requirements of the 3-horsepower and 4-horsepower Rankine engines were larger than the amount available from the solar system. The largest Rankine engine forced the collectors to operate at the lowest possible temperature allowed by the controller, and the

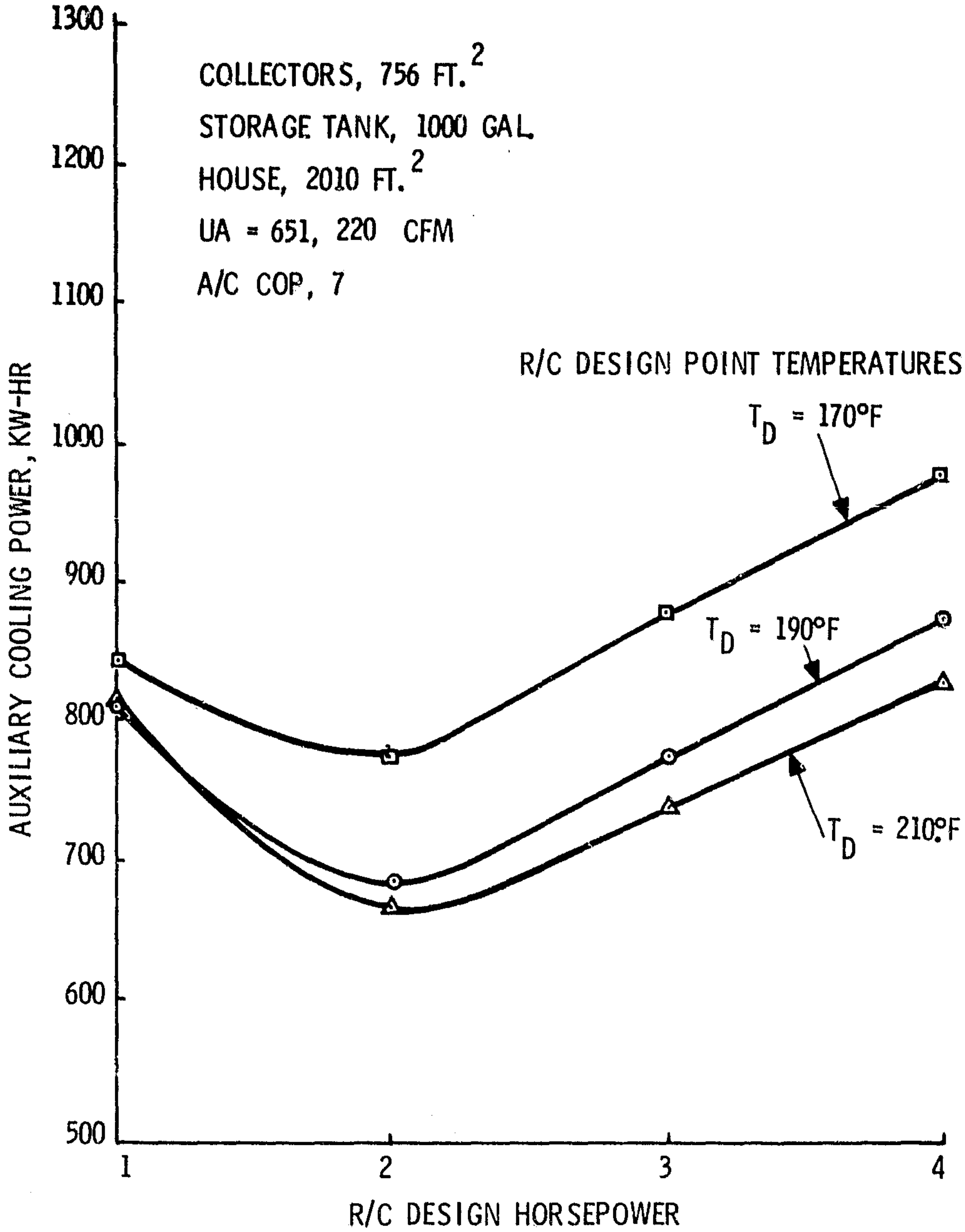

Figure 4-47. Auxiliary Cooling Power versus R/C Design Horsepower at Varying Design Temperatures - 3-Ton Unit

improvement in the solar system efficiency did not offset the decrease in the engine thermal efficiency. Thus, the overall operating efficiency of the large engine system decreased and its auxiliary power requirements increased. The smallest engine considered, the 1-horsepower engine, proved to be undersized for the cooling loads considered. From the results of these simulations, it is apparent that the 2-horsepower engine output was more evenly matched to the power requirements of the air conditioner a greater percentage of the time, with a minimum electrical power expended in providing the added system capacity. Another parameter which was investigated was the effect of the design temperature on the cooling performance of the Rankine engine. As expected, the engine performance improved with increasing temperatures, but this improvement was insufficient to overcome the degradation in collector efficiency. Figure 4-48 illustrates the effect of temperature on various system efficiencies. As the operational temperature drops below 200°F, the combined efficiency ($\eta_{coll} \times \eta_{RC}$) for the 190°F and 210°F designs shows a minimal variation. Consequently, the difference in performance becomes so small that the 190°F design point was selected as the preferred choice. These combined results also indicate that for the location and weather data analyzed, the 210°F design is operating in the lower temperature range a great percentage of the time. The selection of 2 horsepower as a reasonable optimum was based on a Compressor COP of 7.0. But data from presently available components indicated a more realistic value of 6.0, which increased the baseline Rankine-engine size to 2.36 horsepower for the 3-ton cooling system.

The results from the variable-speed simulation showed a net 2.6 percent yearly savings in auxiliary energy consumption over a similar constant-speed design in Atlanta. The selection of a different site with different cooling requirements and solar input can have a large effect on the choice of the best control option and may provide a greater savings for the variable-speed control.

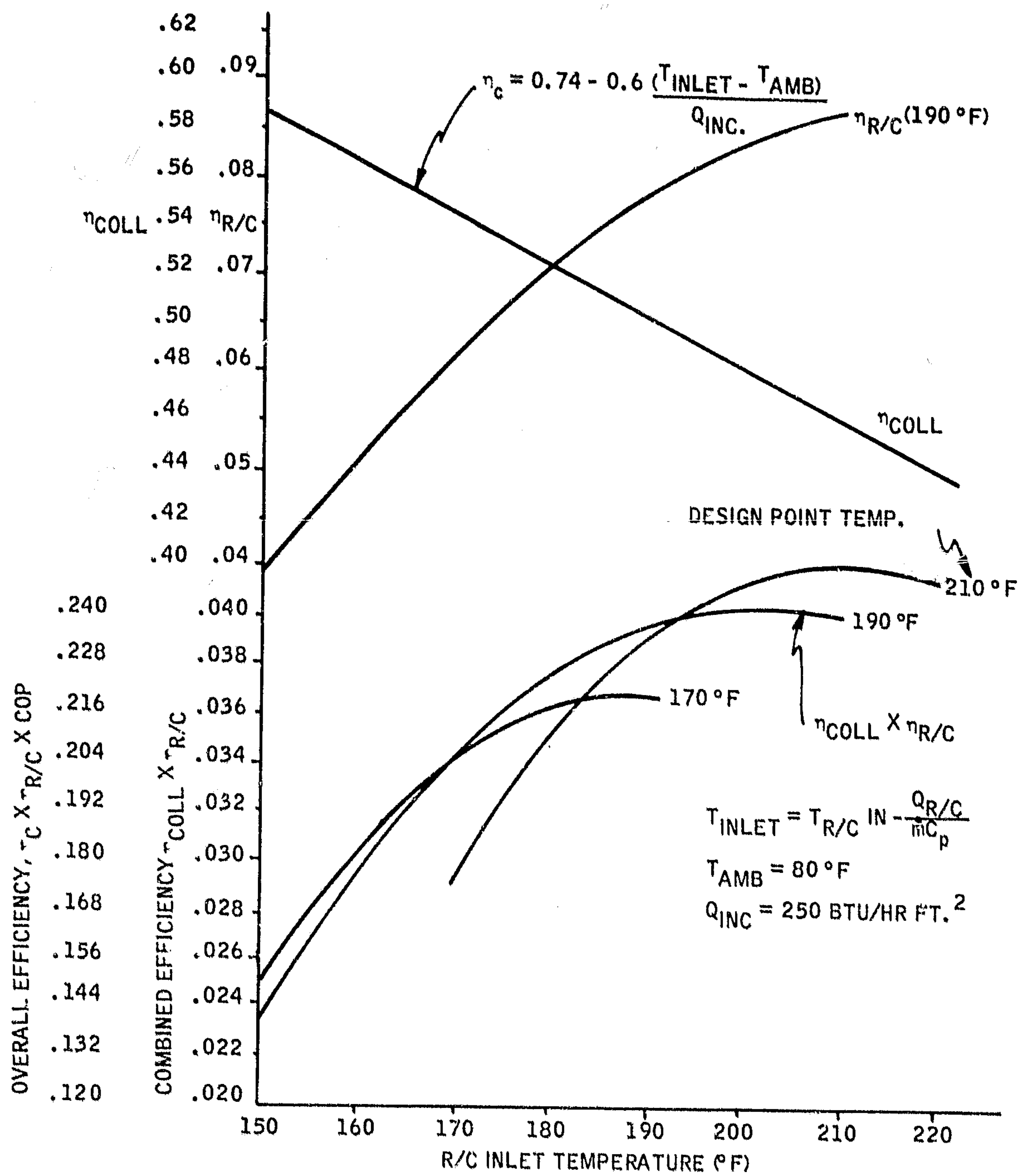

Figure 4-48. Effect of Inlet Temperature versus System Efficiency

The simulation study for the 25-ton unit was completed for both R-113 and R-11 as working fluids. Figure 4-49 illustrates the results for the various engine sizes considered. Since R-113 shows improved efficiency with an approximately 8 percent decrease in auxiliary energy consumption, this fluid was chosen to complete the optimization study.

The selection of the optimum design based on the results illustrated in Figure 4-50 was similar to the 3-ton unit. Rankine-engine design powers between 15 and 20 horsepower showed no major effect on the auxiliary energy consumption for the 190°F and 210°F designs. Further, the total auxiliary energy consumed was low and, in particular, the differences between the various power levels were very small. Thus, the selected design power level was the value required to drive the air conditioner at its design load. For a 25-ton air conditioner with a Compressor COP of 6.0, a 19.6-horsepower engine was chosen.

In summary, the simulation results were used to select the proper design temperature and power levels for each of the system capacities considered. With present plans to provide multiple 25-ton units to supply the cooling requirements of the commercial building, the following designs were selected:

- 3-Ton - Constant Speed:
  - T_D = 190°F control range (150°F to 200°F)
  - Horsepower at 190°F = 2.36
  - R-113 working fluid
- 25-Ton - Constant Speed:
  - T_D = 190°F control range (150°F to 200°F)
  - Horsepower at 190°F = 19.6
  - R-113 working fluid

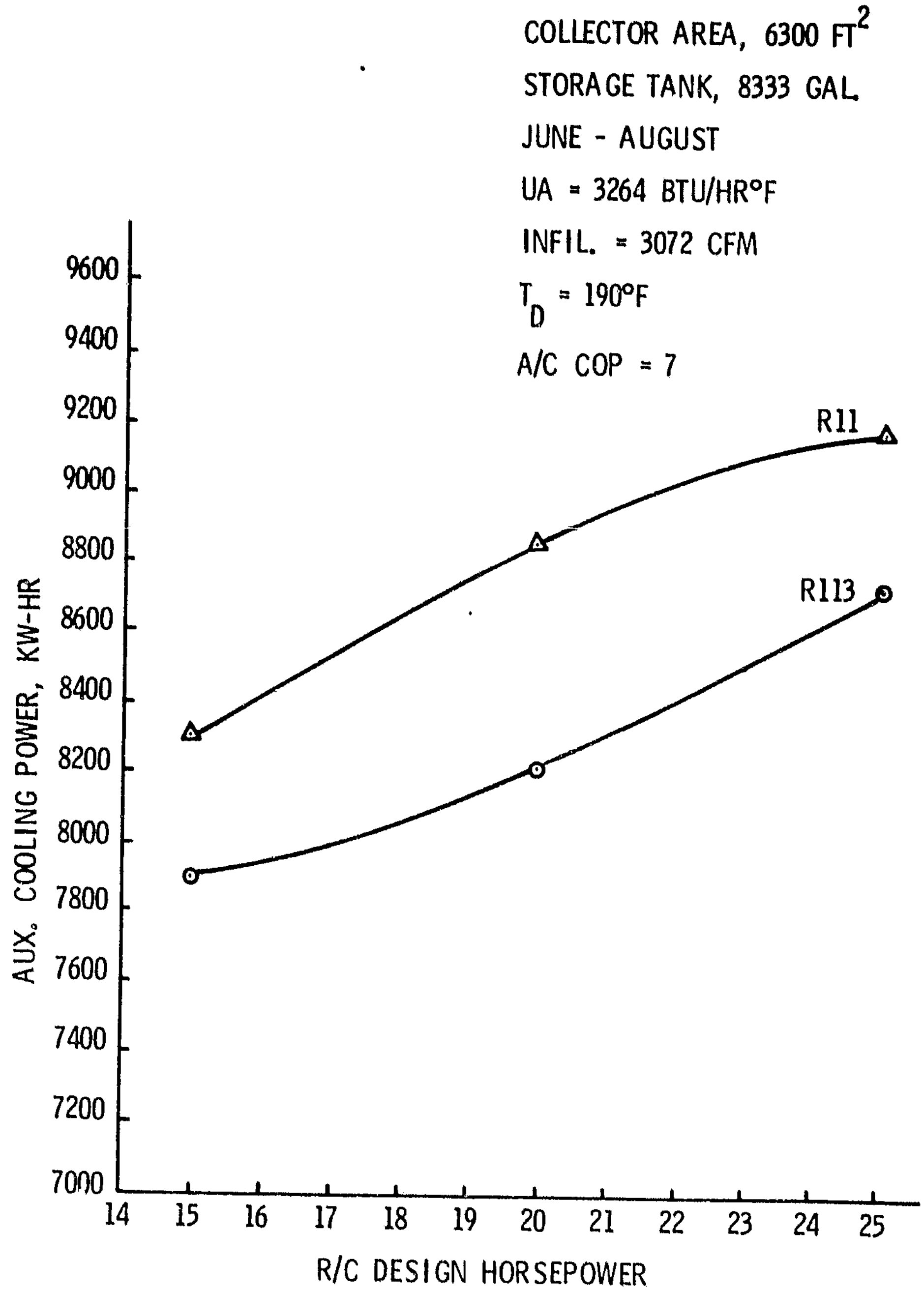

Figure 4-49. Refrigerant Effect on Rankine-Cycle Design Horsepower versus Auxiliary Cooling Power for Atlanta Multifamily Residence

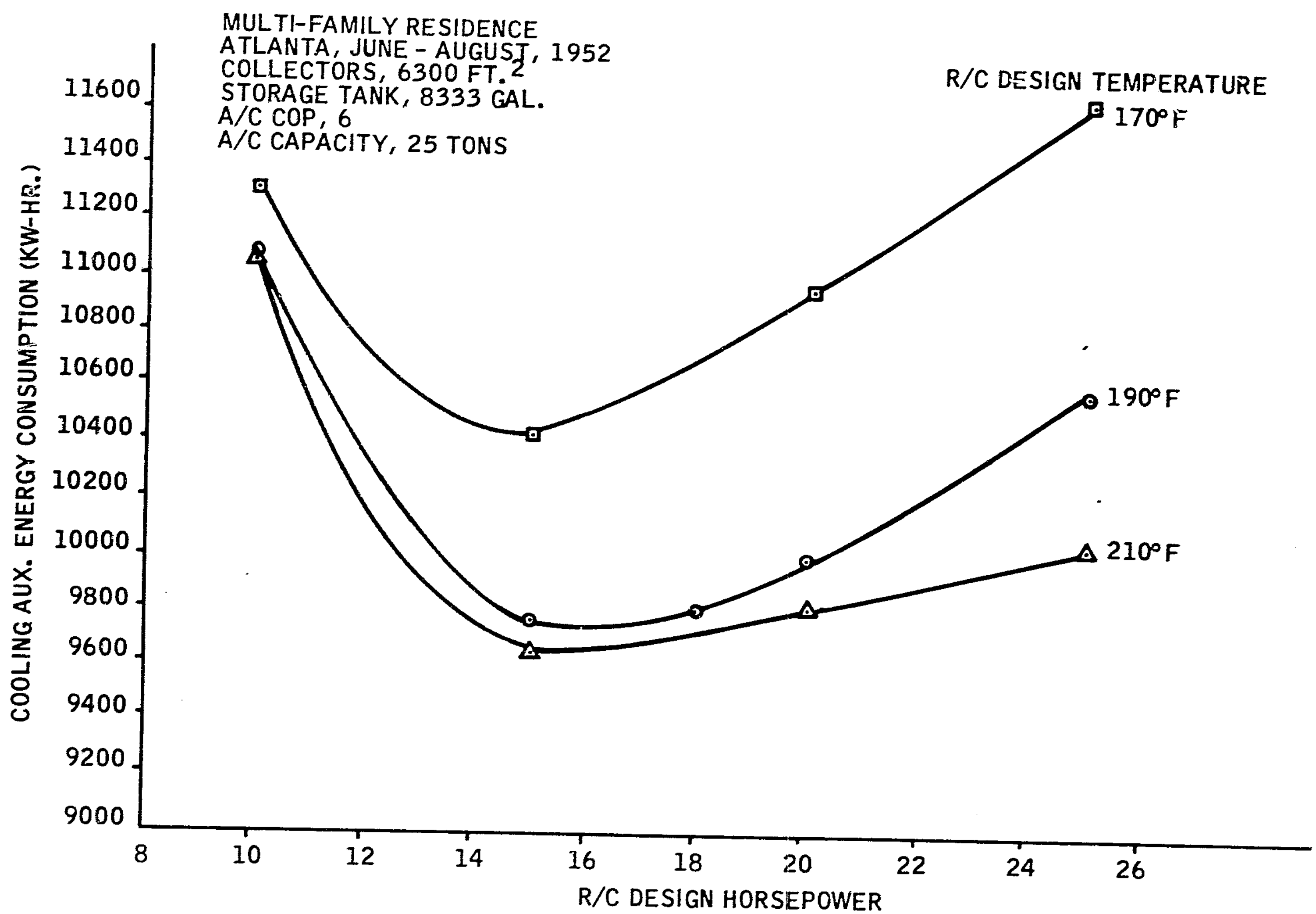

Figure 4-50. Auxiliary Cooling Power versus R/C Design Horsepower at Varying Design Temperatures - 25-Ton Unit

4.13 HEATING SYSTEMS

The following subsections delineate the baseline heating systems. During analysis of the Heating/Cooling systems it was determined that the previously designed heating systems for multi-family and commercial systems should be redesigned to incorporate the same features as the Heating/Cooling systems. This has distinct advantages in modularity of design and better freeze protection for the heating systems.

The proposed solar-assisted heating systems are liquid solar collector systems interfacing with conventional fossil fuel-fired furnaces. These proposed system designs may be easily sized to fit a wide spectrum of applications. First, the collectors are modular and can be combined in arrays to satisfy site-specific collector configurations. Second, the auxiliary subsystems are selected from a broad product line of fossil fuel-fired warm air furnaces, allowing many choices to fit site-specific requirements. Finally, the storage, hot water, transport and control subsystems are commercially produced items in a broad range of sizes. This subsystem modularity allows variations in system design to accommodate the variable performance requirements that are expected nationwide.

The proposed solar systems are designed to maximize the amount of solar energy collected for use and storage. This is accomplished by:

- A control system that minimizes collector inlet temperatures (maximizes energy into the building)
- Optimum transfer rate heat exchangers
- Direct collector to space heating by by-passing storage
- Using high-performance flat plate collectors
- Storage which can be operated in parallel and independently from collector loop

System reliability and maintainability is assured through design features which include the following:

- A closed collector loop for over temperature protection via a purge coil
- A control system employing simple logic
- A minimum of components in the system
- Manifolding external to the collector modules

Maintenance of space temperature, hence occupant comfort, is assured through the use of the following components and design techniques:

- A two-stage thermostat with a minimum differential for solar operation
- Conventional furnace control of air temperature to the space

The systems have been designed to minimize contamination of the potable water supply by the use of:

- A two-fluid loop system that isolates the collector heat transfer fluid
- A system in which domestic water pressure is higher than system pressures

4.13.1 Single-Family Residential Heating System Description

The proposed system for a single-family residential heating system is a single-loop, solar-assisted, hydronic-to-warm air heating subsystem with

solar-assisted domestic water heating. The system is composed of the following major components:

- Liquid cooled flat plate collectors
- A water storage tank
- A passive solar fired domestic water preheater
- A gas-fired hot-water heater
- A gas-fired warm-air furnace with hot-water coil unit
- A tube-and-shell heat exchanger, three pumps, and associated pipes and valving
- A control system
- An air-cooled heat purge unit

The arrangement of components within the system is as shown in Figure 4-51. The system consists of a glycol/water collector loop which interfaces with a water storage/heating loop, through a tube-and-shell heat exchanger. A domestic hot-water preheat coil is located in the storage tank.

The glycol/water collector loop consists of the solar collectors, the shell side of the heat exchanger, the collector loop pump P_1 and the purge coil. The water storage/heating loop consists of the storage tank, the storage/heating pump P_2, the tube side of the heat exchanger, heating pump P_3, the heating coil and a control valve for the required modes of operation.

The system provides six modes of operation:

- Direct heating from collectors
- Direct heating from storage
- Domestic hot water preheater

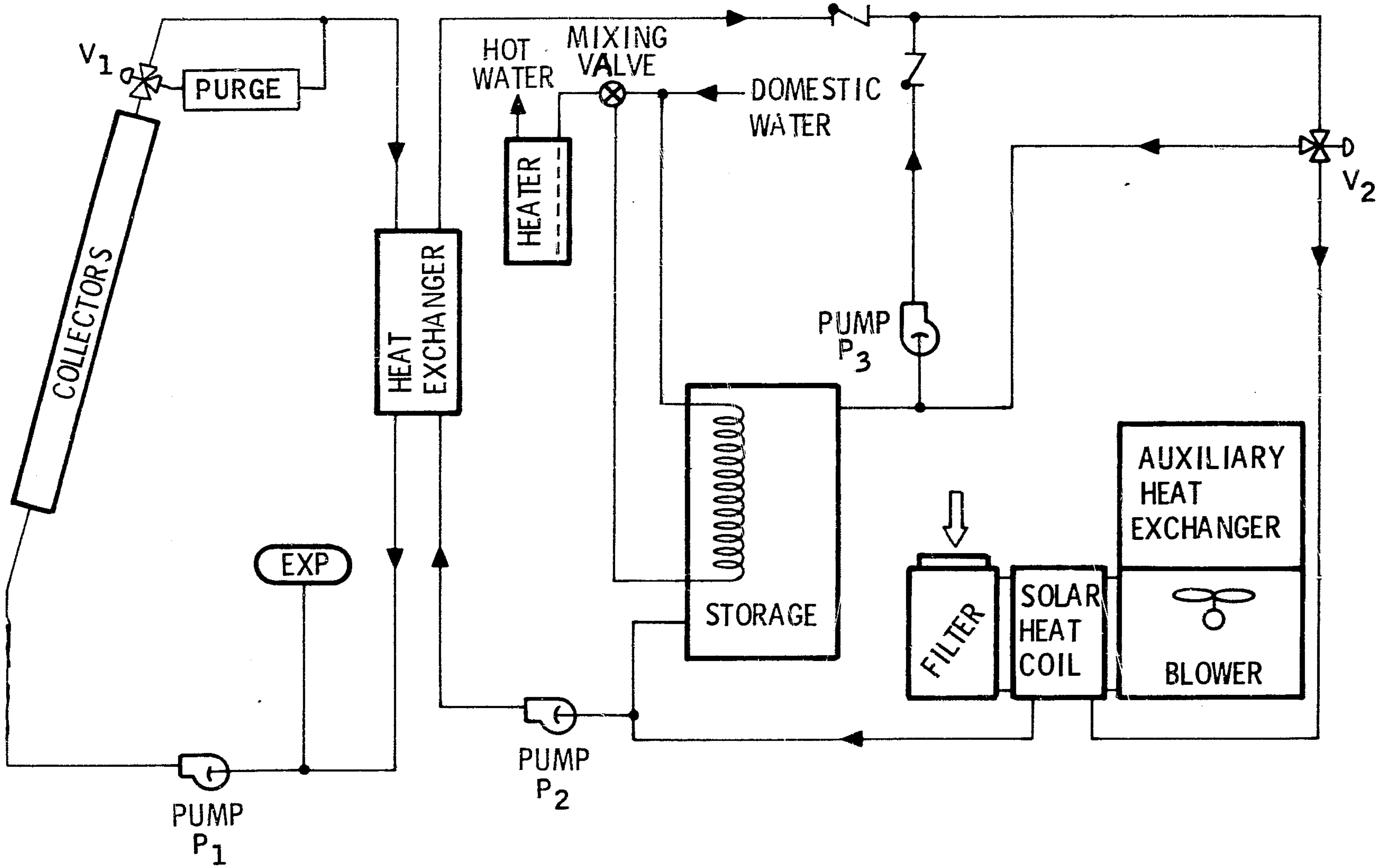

Figure 4-51. Single-Family Residential Heating System Schematic

- Storage charging
- Auxiliary heating (insufficient solar)
- Purge excess energy

4.13.1.1 Heating Subsystem Operation -- When solar energy is available and heating is required, the collectors supply heat directly to the system. Pump P_1 provides the heat transfer fluid movement in the collector loop, pump P_2 circulates the storage/heating loop, and the furnace blower moves the space air over the heat coil.

During periods of high solar radiation and low heating demand, both collector and storage loops operate simultaneously, with the storage charge pump P_2 charging the storage tank by removing water from the bottom, adding energy in the heat exchanger and returning it to the top of the storage tank, thus taking advantage of thermal stratification. During periods of high solar radiation and low heating demand and with the storage tank fully charged, the system temperatures will increase and, as an overtemperature protective device, the purge coil operates, controlling the downstream temperatures to a preselected value.

When solar energy is not available and heating is required, storage supplies heat directly to the solar heat coil. Pump P_3 extracts heat from the top of the storage tank and returns it to the bottom, again taking advantage of the tank stratification. If the storage tank temperature is not high enough to supply heating, or the heating load cannot be satisfied by the solar system, the second-stage thermostat activates the auxiliary furnace until a comfortable temperature is maintained.

4.13.1.2 Specific Subsystem/Component Design Criteria -- The location of each of the components within the system is determined by the following criteria:

- Optimize performance of entire system
- Provide reliable operation of each component
- Provide control functions for each mode of operation

The collector loop pump P_1 is located downstream of the heat exchanger ahead of the solar collectors. This location enables the pump to handle the glycol/water fluid at the lowest operating temperatures in the collector loop. The expansion tank is located at the pump suction to provide system pressure control and air elimination for proper system operation.

When the system is in the storage charging mode, valve V_2 diverts flow around the solar heat coil. When the system is in the purge mode, control valve V_1 diverts flow through the purge coil unit which is placed downstream of the solar collectors for protection of other system components from overtemperature.

In the water storage loop, the storage/heating pump P_2 is located so as to provide a counter-flow arrangement at the storage heat exchanger. Pumps P_2 and P_3 are used to take advantage of stratification of storage temperature within the tank. Pump P_3 draws hotter water from the top of the storage tank for heating from storage mode of operation. Pump P_2 draws cooler water from the bottom of the storage tank for storage charging and direct heating modes of operation.

The solar domestic hot-water preheat coil, in combination with the storage heat exchanger, provides a double-wall separation between the ethylene glycol/water solution in the collector loop and the potable domestic hot-water system.

A thermostatic mixing valve keeps the temperature below 140°F. A conventional water heater is available downstream to add auxiliary energy as required to obtain 140°F water.

4.13.2 Multiple-Family Residential Heating System Description

The proposed system for a multiple-family residential heating and cooling system is a single-loop, solar-powered, two-pipe hydronic heating system with a separate water storage and domestic hot-water heating loop. The central heating is provided by either direct or stored solar energy with each individual zone having auxiliary heating available. The system is composed of the following major components:

- Liquid-cooled flat-plate collectors
- A water storage tank
- A solar-fired domestic water preheater with storage water tube bundle and circulating pump
- A gas-fired hot-water heater
- A tube-and-shell heat exchanger, four central system pumps, and associated piping and valving
- A control system
- An air-cooled heat purge unit with fan
- Twelve individual subsystems consisting of a circulating pump and a gas-fired warm-air furnace with a water heating coil

The arrangement of components within the system is shown in Figure 4-52. The system consists of a glycol/water collector loop which interfaces with a water storage loop, through a tube-and-shell heat exchanger.

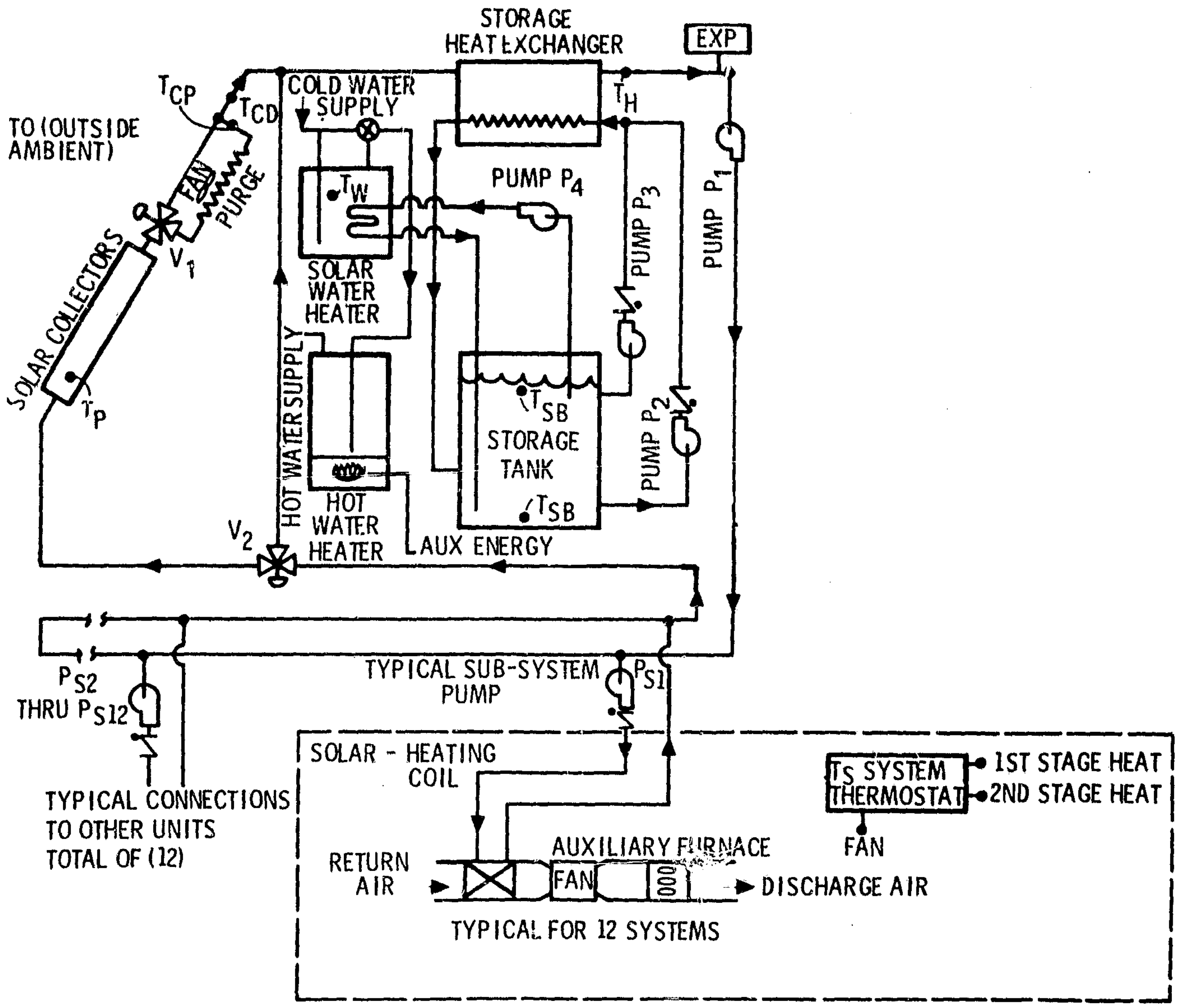

Figure 4-52. Multiple-Family Residential Heating System

The glycol/water collector loop consists of the solar collectors, the shell side of the storage heat exchanger, the heating coil and pump P_1, the purge coil, and two control valves as required for the different modes of operation.

The water storage loop consists of the storage tank, two storage pumps, P_2 and P_3, and the tube side of the storage heat exchanger, and the solar DHW preheater pump, P_4.

The system provides seven modes of operation:

- Direct heating from collectors
- Direct heating from storage
- Direct heating and storage simultaneously
- Auxiliary heating (insufficient solar)
- Storage charging
- Domestic hot-water preheater
- Purge excess energy

4.13.2.1 Heating Subsystem Operation -- When solar energy is available and heating is required, the collectors supply heat directly to the system. Pump P_1 provides the heat transfer fluid movement in the primary loop, the subsystem pump circulates the secondary loop, and the furnace blower moves the space air over the heat coil.

During periods of high solar radiation and low heating demand, both heating and storage loop operate simultaneously, with the storage charge pump charging the storage tank by removing water from the bottom, adding energy in the heat exchanger and returning it to the center of the storage tank, thus taking advantage of thermal stratification. During periods of high solar radiation and low heating demand and with the storage tank fully charged, the system temperatures will increase and, as an overtemperature protective device, the purge coil operates, controlling the downstream temperatures to a preselected value.

When solar energy is not available and heating is required, storage supplies heat to the space through the heat exchanger. Pump P_1 drives the primary loop and pump P_3 extracts heat from the top of the storage tank and returns it to the center, again taking advantage of the tank stratification. If the storage tank temperature is not high enough to supply heating, the heating load cannot be satisfied by the solar system and the second-stage thermostat activates the auxiliary furnace until a comfortable temperature is maintained.

4.13.2.2 Specific Subsystem/Component Design Criteria -- The location of each of the components within the system is determined by the following criteria:

- Optimize performance of entire system
- Provide reliable operation of each component
- Provide control functions for each mode of operation

The primary loop expansion tank is located at the suction side of pump P_1. This provides for air elimination and system pressure control during all modes of operation.

The storage heat exchanger, within the collector loop, is placed downstream of the solar collectors and ahead of the heating loads so as to provide a means for inlet temperature control. This also provides for a convenient method for simultaneous storage charging.

When the system is heating from storage, control valve V_2 diverts flow around the solar collectors.

When the system is in the purge mode, control valve V_1 diverts flow through the purge coil unit which is placed downstream of the solar collectors for protection of other system components from overtemperature.

In the water storage loop, the storage pumps P_2 and P_3 are located so as to provide a counter-flow arrangement at the storage heat exchanger. Two pumps are used to take advantage of stratification of storage temperature within the tank. The common return line from the storage heat exchanger enters the middle of the storage tank. Pump P_3 draws hotter water from the top of the storage tank for the heating from storage mode of operation. Pump P_2 draws cooler water from the bottom of the storage tank for storage charging of collected solar energy.

The solar domestic hot-water preheater, in combination with the storage heat exchanger, provides a double-wall separation between the ethylene glycol/water solution in the collector loop and the potable domestic hot-water system.

The solar domestic hot-water preheater heats incoming cold water with solar-heated storage water through a tube bundle circulated by pump P_4. Storage water is drawn from the top of the storage tank and returned to the bottom to take advantage of thermal stratification.

A thermostatic mixing valve keeps the temperature below 140°F. A conventional water heater is available downstream to add auxiliary energy as required to obtain 140°F water.

4.13.3 Commercial Building Heating System Description

The proposed system for a commercial heating and cooling system is a single-loop, solar-powered, two-pipe hydronic heating system with a separate water storage and domestic hot-water heating loop. The central heating is provided by either direct or stored solar energy and each zone has auxiliary heating available.

The system is composed of the following major components:

- Liquid-cooled flat-plate collectors
- A water storage tank
- A solar-fired domestic water preheater with storage water tube bundle and circulating pump
- A gas-fired hot-water heater
- A tube-and-shell heat exchanger, four central system pumps, and associated piping and valving
- A control system
- An air-cooled heat purge unit with fan
- Four individual subsystems consisting of a circulating pump and a gas-fired warm-air furnace with a water heating coil

The arrangement of components within the system is shown in Figure 4-53. The system consists of a glycol/water collector loop which interfaces with a water storage loop, through a tube-and-shell heat exchanger.

The glycol/water collector loop consists of the solar collectors, the shell side of the storage heat exchanger, the heating coil and pump P_1, the purge coil, and two control valves as required for the different modes of operation.

The water storage loop consists of the storage tank, two storage pumps, P_2 and P_3, and the tube side of the storage heat exchanger, and the solar DHW preheater pump, P_4.

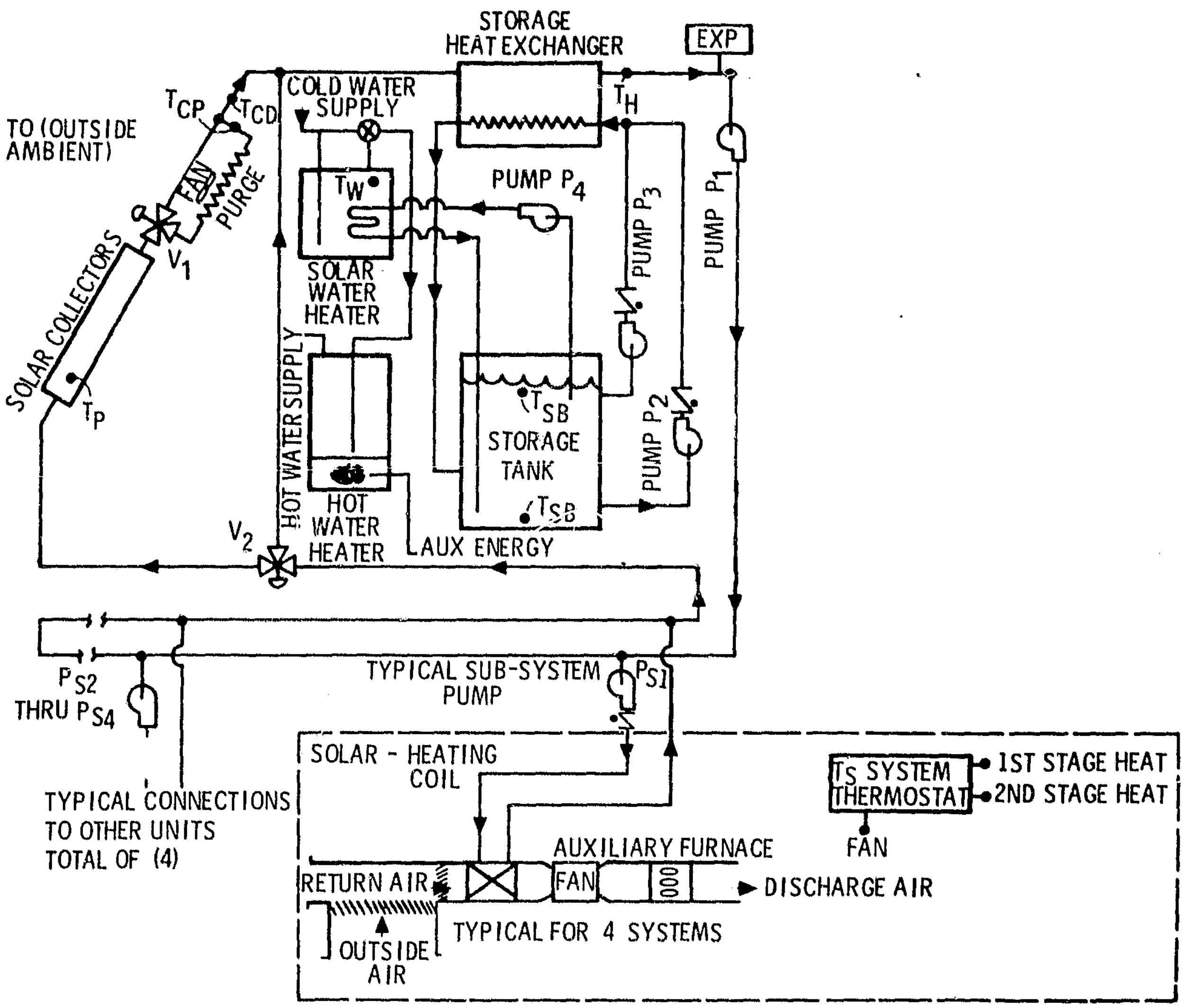

Figure 4-53. Commercial Building Heating System

The system provides seven modes of operation:

- Direct heating from collectors
- Direct heating from storage
- Direct heating and storage simultaneously
- Auxiliary heating (insufficient solar)
- Storage charging
- Domestic hot-water preheater
- Purge excess energy

4.13.3.1 <u>Heating Subsystem Operation</u> -- When solar energy is available and heating is required, the collectors supply heat directly to the system. Pump P_1 provides the heat transfer fluid movement in the primary loop, the subsystem pump circulates the secondary loop, and the furnace blower moves the space air over the heat coil.

During periods of high solar radiation and low heating demand, both heating and storage loop operate simultaneously, with the storage charge pump charging the storage tank by removing water from the bottom, adding energy in the heat exchanger and returning it to the center of the storage tank, thus taking advantage of thermal stratification. During periods of high solar radiation and low heating demand and with the storage tank fully charged, the system temperatures will increase and, as an overtemperature protective device, the purge coil operates, controlling the downstream temperatures to a preselected value.

When solar energy is not available and heating is required, storage supplies heat to the space through the heat exchanger. Pump P_1 drives the primary loop and pump P_3 extracts heat from the top of the storage tank and returns it to the center, again taking advantage of the tank stratification.

If the storage tank temperature is not high enough to supply heating, the heating load cannot be satisfied by the solar system and the second-stage thermostat activates the auxiliary furnace until a comfortable temperature is maintained.

4.13.3.2 Specific Subsystem/Component Design Criteria -- The location of each of the components within the system is determined by the following criteria:

- Optimize performance of entire system
- Provide reliable operation of each component
- Provide control functions for each mode of operation

The primary loop expansion tank is located at the suction side of pump P_1. This provides for air elimination and system pressure control during all modes of operation.

The storage heat exchanger, within the collector loop, is placed downstream of the solar collectors and ahead of the heating loads so as to provide a means for inlet temperature control. This also provides for a convenient method for simultaneous storage charging.

When the system is heating from storage, control valve V_2 diverts flow around the solar collectors.

When the system is in the purge mode, control valve V_1 diverts flow through the purge coil unit which is placed downstream of the solar collectors for protection of other system components from overtemperature.

In the water storage loop, the storage pumps P_2 and P_3 are located so as to provide a counter-flow arrangement at the storage heat exchanger. Two pumps are used to take advantage of stratification of storage temperature

within the tank. The common return line from the storage heat exchanger enters the middle of the storage tank. Pump P_3 draws hotter water from the top of the storage tank for the heating from storage mode of operation. Pump P_2 draws cooler water from the bottom of the storage tank for storage charging of collected solar energy.

The solar domestic hot-water preheater, in combination with the storage heat exchanger, provides a double-wall separation between the ethylene glycol/water solution in the collector loop and the potable domestic hot-water system.

The solar domestic hot-water preheater heats incoming cold water with solar-heated storage water through a tube bundle circulated by pump P_4. Storage water is drawn from the top of the storage tank and returned to the bottom to take advantage of thermal stratification.

A thermostatic mixing valve keeps the temperature below 140°F. A conventional water heater is available downstream to add auxiliary energy as required to obtain 140°F water.

4.14 HEATING AND COOLING SYSTEMS

The following subsections delineate the baselines for Heating and Cooling Systems and the analytical study results giving the features of each system.

4.14.1 General Baseline System Description

The proposed solar-assisted heating and cooling systems are single-loop solar collector systems interfacing with a conventional fossil-fuel-fired furnace and a Rankine-cycle airconditioning subsystem. These proposed heating subsystem designs may be easily sized to fit a wide spectrum of

applications. First, the collectors are modular and can be combined in arrays to satisfy site specific collector configurations. Second, the auxiliary subsystems are selected from a broad product line of fossil-fuel-fired warm-air furnaces, allowing many choices to fit site specific requirements. Third, the storage, hot-water, transport, and control subsystems are commercially produced items in a broad range of sizes. This subsystem modularity allows variations in system design to accommodate the variable performance requirements that are expected nationwide.

The Rankine-driven cooling subsystem will be designed and developed in two sizes: 2-ton and 25-ton. Lennox marketing analysis was used to select the most popular residential size on a nationwide basis. The 25-ton size represents the best size adaptable to larger multiple-family and commercial systems as individual or multiple systems.

The proposed solar systems are designed to maximize the amount of solar energy collected for use and storage. This is done by:

- Using high-performance flat-plate collectors
- A control system that minimizes collector inlet temperatures (maximizes energy into the building by increasing collector efficiency) and provides flexible multi-loop operation
- Independent heat exchangers using optimum transfer rate
- Direct collector to space heating, bypassing storage
- Direct collector to Rankine cooling, bypassing storage
- Storage which can be operated in parallel and independently from heating or cooling loop

System reliability and maintainability are assured through design features which indicate the following:

- A purge subsystem located to protect downstream components from overtemperatures
- A control system using simple logic
- A minimum of components in the system
- Manifolding external to the collector modules

Maintenance of space temperature, hence occupant comfort, is assured through the use of the following components and design techniques:

- A two-stage thermostat with a minimum differential for solar operation and auxiliary backup for heating and cooling
- Conventional furnace control of air temperature to the space.

The systems have been designed to minimize contamination of the potable water supply by the use of:

- A two-fluid-loop storage system that isolates the collector heat transfer fluid by two heat exchangers (two walls)
- A system in which domestic water pressure is higher than system pressures, thus assuring early detection of leaks.

4.14.2 Single-Family Residential Heating and Cooling System Description

The proposed system for a single-family residential heating and cooling system is a single-loop, solar-assisted, hydronic-to-warm air heating subsystem with solar-assisted domestic water heating and a Rankine-driven expansion air-conditioning subsystem. The system is composed of the following major components:

- Liquid cooled flat plate collectors
- A water storage tank
- A passive solar fired domestic water preheater
- A gas-fired hot-water heater
- A gas-fired warm-air furnace with hot-water coil unit
- A Rankine-driven direct-expansion air conditioner with auxiliary electric motor
- Water pump and cooling tower
- A tube-and-shell heat exchanger, four pumps, and associated pipes and valving
- A control system
- An air-cooled heat purge unit

The arrangements of components within the system is as shown in Figure 4-54. The system consists of a glycol/water collector loop which interfaces with a water storage loop, through a tube-and-shell heat exchanger. A domestic hot-water preheat coil is located in the storage tank.

The glycol/water collector loop consists of the solar collectors, the shell side of the storage heat exchanger, the heating coil and pump P_2, the Rankine boiler and pump P_1, the purge coil, and three control valves as required for the different modes of operation.

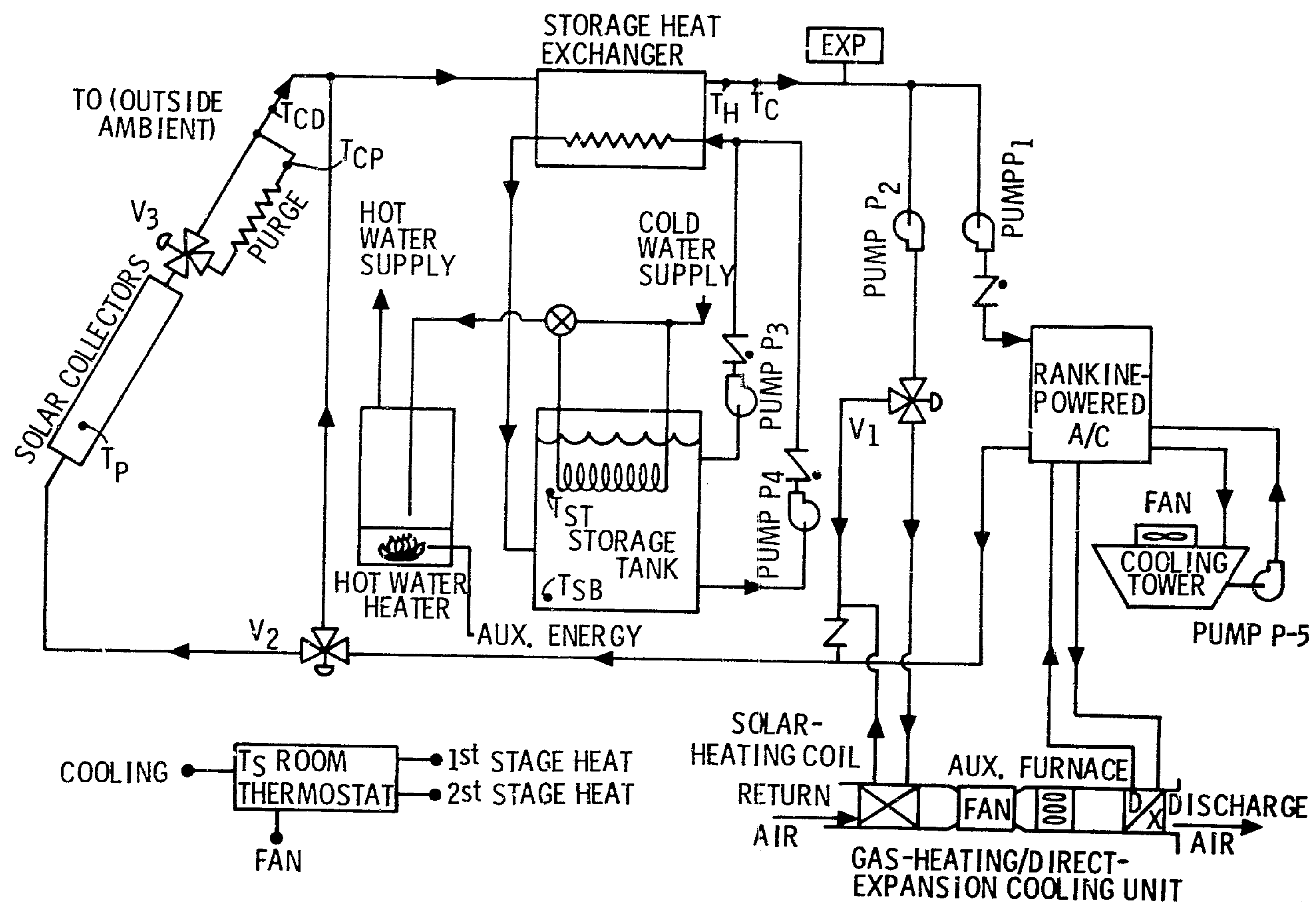

Figure 4-54. Single-Family Residential H/C System

The water storage loop consists of the storage tank, storage pumps P_3 and P_4, and the tube side of the storage heat exchanger.

The system provides 10 modes of operation:

- Direct heating from collectors
- Direct heating from storage
- Direct heating and storage simultaneously
- Auxiliary heating (insufficient solar)
- Rankine cooling from collectors
- Rankine cooling from storage
- Rankine cooling and storage simultaneously
- Electric motor auxiliary cooling
- Domestic hot water preheater
- Purge excess energy

4.14.2.1 Heating Subsystem Operation -- When solar energy is available and heating is required, the collectors supply heat directly to the furnace through the hot-water coil in the return air duct. Pump (P_2) provides the heat-transfer fluid movement in this loop and the furnace blower moves the building air through the heat coil. When the heating demand is satisfied, valve V_1 diverts the fluid around the hot-water coil and pump P_4 operates, charging the storage tank by removing water from the bottom, adding energy in the heat exchanger and returning it to the center of the storage tank, thus taking advantage of stratification. During high solar radiation and low heating demand, both heating and storage loop operate simultaneously. If additional energy is still available, the purge coil operates, controlling the downstream temperatures to a preselected value.

When solar energy is not available and heating is required, storage supplies heat to the furnace through the heat exchanger. Pump P_2 drives the outside loop and pump P_3 extracts heat from the top of the storage tank and returns it to the center, again taking advantage of the tank stratification. If the storage tank temperature is not high enough to supply heating, the second-stage thermostat activates the auxiliary furnace until a comfortable temperature is maintained.

4.14.2.2 Cooling Subsystem Operation -- When solar energy is available and cooling is required, the collectors supply heat directly to the Rankine boiler. Pump P_1 provides the heat-transfer fluid movement and, if necessary, can be sized differently than P_2 to improve efficiency of the collector and Rankine operation. The Rankine drives a high Coefficient-of-Performance (COP) compressor which, in turn, provides conventional direct-expansion cooling. When the cooling demand is satisfied, pump P_2 is shut down and the system reverts to the storage mode explained in the heating subsystem.

During high solar radiation and low cooling demand, simultaneous cooling and storage is available, using pump P_1 and pump P_4. If additional energy is still available, the purge coil operates by controlling the downstream temperatures to a preselected value. This is an infrequent mode and would occur if coils are oversized for a large heating demand.

When solar energy is not available or insufficient to operate the Rankine engine at the design horsepower (2.36 horsepower at 190°F collector outlet), an electric motor will operate the air conditioner independently or to make up the difference between the required horsepower and that supplied by the R/C. Storage is used to supply energy in the same manner as in the heating subsystem, except that pump P_1 is the prime mover in the glycol/water loop. The baseline design uses a constant-speed compressor and therefore the electric motor is on-line at all times, supplying the balance of the required horsepower.

A variation of this concept is under study which uses a variable-speed R/C input to the compressor and thus variable-cooling output. The auxiliary motor operates only when the second-stage thermostat indicates the cooling load is not satisfied. In second stage, the system operates as a constant-speed system, delivering 3 tons of cooling.

4.14.2.3 Specific Subsystem/Component Considerations -- The heating/storage circuit pump, P_2, and the cooling circuit pump, P_1, are located ahead of the loads so as to provide a common location for the expansion tank. This provides for air elimination and system pressure control during all modes of operation.

When the system is heating or cooling from storage, control valve V_2 diverts flow around the solar collectors.

When the system is in the purge mode, control valve V_3 diverts flow through the purge coil which is placed downstream of the solar collectors for protection of other system components from overtemperature.

In the water storage loop, the storage pumps, P_3 and P_4, are located so as to provide a counter-flow arrangement at the storage heat exchanger. Two pumps are used to take advantage of stratification of storage temperature within the tank. The common return line from the storage heat exchanger enters the middle of the storage tank. Pump P_3 draws hot water from the top of the storage tank for heating or cooling from storage. Pump P_4 draws cooler water from the bottom of the storage tank for storage charging.

The domestic hot-water preheat coil, in combination with the storage heat exchanger, provides a double-wall separation between the ethylene glycol/water solution in the collector loop and the potable domestic hot-water system.

The domestic hot-water coil in the storage tank transfers energy to the entering cold water. A thermostatic mixing valve keeps the temperature below 140°F. A conventional water heater is available downstream to add auxiliary energy as required to obtain 140°F water.

The location of each of the components within the systems is determined by the following criteria:

- Optimize performance of entire system.
- Provide reliable operation of each component.
- Provide control functions for each mode of operation.

The storage heat exchanger, within the collector loop, is placed downstream of the solar collectors and ahead of the loads so as to provide a means for inlet temperature control to the loads. This also provides for a convenient method for simultaneous storage charging.

The heating/storage circuit and the cooling circuit are placed in parallel, downstream of the storage heat exchanger. Each circuit is controlled by its pump and protected from backflow by a check valve.

4.14.3 Multiple-Family Residential Heating and Cooling System Description

The proposed system for a multiple-family residential heating and cooling system is a single-loop, solar-powered, two-pipe hydronic heating and cooling system with a separate water storage and domestic hot-water heating loop. Being a two-pipe design, the central system is either in the heating mode or the cooling mode as determined by a Cooling Load Analyzer. The central heating is provided by either direct or stored solar energy and the central cooling is provided by a solar-powered Rankine engine/auxiliary electric motor-driven water chiller. The system is composed of the following major components:

- Liquid-cooled flat-plate collectors
- A water storage tank
- A solar-fired domestic water preheater with storage water tube bundle and circulating pump
- A gas-fired hot-water heater
- A water chiller driven by a Rankine engine and/or auxiliary electric motor
- A tube-and-shell heat exchanger, four central system pumps, and associated piping and valving
- A condenser water pump and cooling tower
- A control system
- An air-cooled heat purge unit with fan
- Twelve individual subsystems consisting of a circulating pump and a gas-fired warm-air furnace with a water heating/cooling coil

The arrangement of components within the system is shown in Figure 4-55. The system consists of a glycol/water collector loop which interfaces with a water storage loop, through a tube-and-shell heat exchanger.

The glycol/water collector loop consists of the solar collectors, the shell side of the storage heat exchanger, the heating coil and pump P_2, the Rankine boiler and pump P_1, the purge coil, and three control valves as required for the different modes of operation.

The water storage loop consists of the storage tank, two storage pumps, P_3 and P_4, and the tube side of the storage heat exchanger, and the solar DHW preheater pump, P_6.

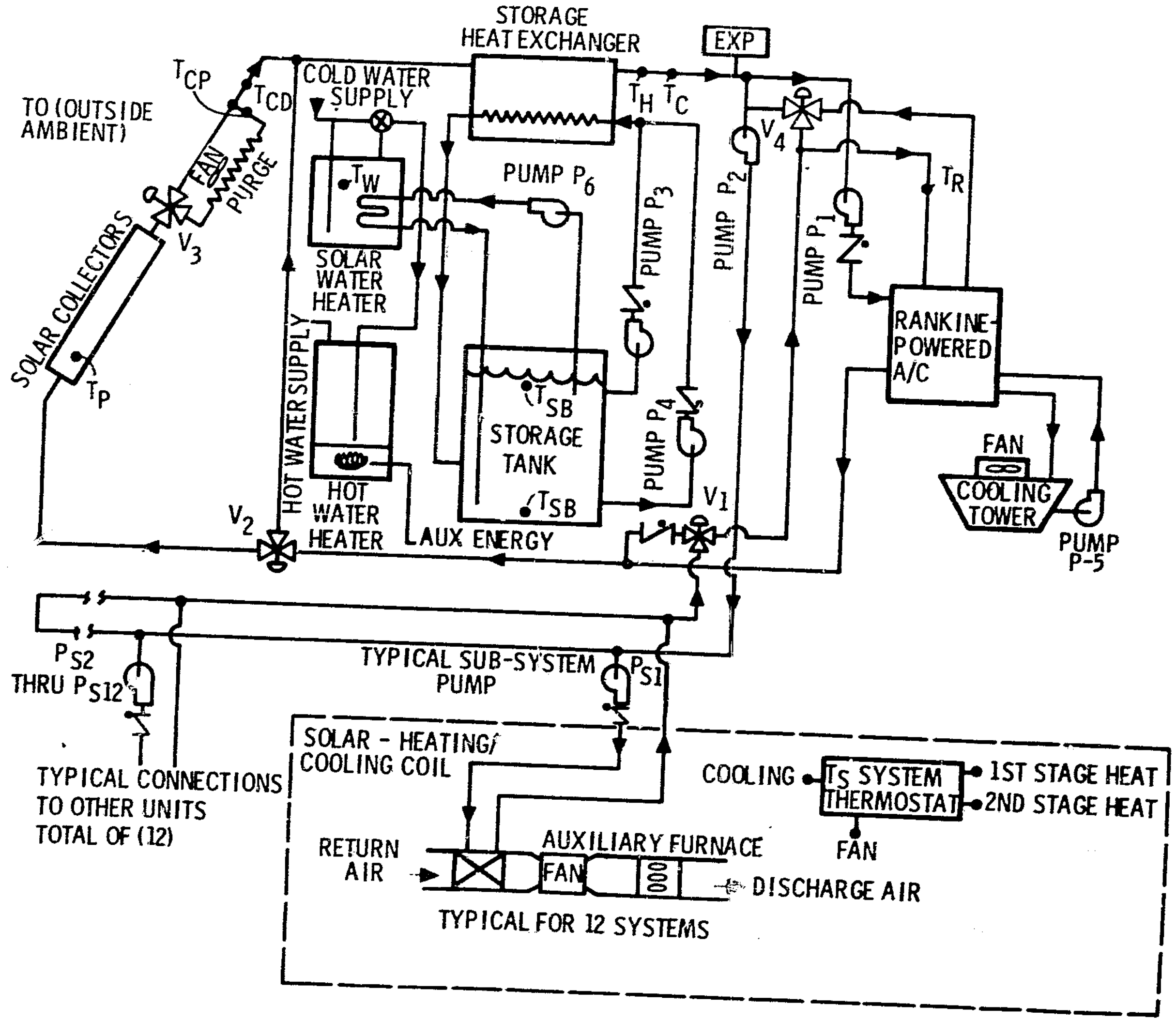

Figure 4-55. Multiple-Family Residential H/C System

The system provides 10 modes of operation:

- Direct heating from collectors
- Direct heating from storage
- Direct heating and storage simultaneously
- Auxiliary heating (insufficient solar)
- Rankine cooling from collectors
- Rankine cooling from storage
- Rankine cooling and storage simultaneously
- Electric motor auxiliary cooling
- Domestic hot-water preheater
- Purge excess energy

4.14.3.1 Heating Subsystem Operation -- When solar energy is available and heating is required, the collectors supply heat directly to the system. Pump P_1 provides the heat transfer fluid movement in the primary loop, the subsystem pump circulates the secondary loop, and the furnace blower moves the space air over the heat coil.

During periods of high solar radiation and low heating demand, both heating and storage loop operate simultaneously, with the storage charge pump charging the storage tank by removing water from the bottom, adding energy in the heat exchanger and returning it to the center of the storage tank, thus taking advantage of thermal stratification. During periods of high solar radiation and low heating demand and with the storage tank fully charged, the system temperatures will increase and, as an overtemperature protective device, the purge coil operates, controlling the downstream temperatures to a preselected value.

When solar energy is not available and heating is required, storage supplies heat to the space through the heat exchanger. Pump P_2 drives the primary loop and pump P_3 extracts heat from the top of the storage tank and returns it to the center, again taking advantage of the tank stratification. If the storage tank temperature is not high enough to supply heating, the heating load cannot be satisfied by the solar system and the second-stage thermostat activates the auxiliary furnace until a comfortable temperature is maintained.

4.14.3.2 Cooling Subsystem Operation -- When solar energy is available and cooling is required, the collectors supply heat directly to the Rankine boiler. Pump P_1 provides the heat transfer fluid movement and, if necessary, can be sized differently than P_2 to improve efficiency of the collector and Rankine engine operation. The Rankine engine drives a high COP water chiller, which, in turn, provides chilled water to the system. When the cooling demand is satisfied, pump P_2 is shut down and the system reverts to the storage mode explained in the heating subsystem. During high solar radiation and low cooling demand, simultaneous cooling and storage is available. The purge coil unit will protect the system from excessive temperatures.

When solar energy is not available or insufficient to operate the Rankine engine at the design horsepower (20 horsepower at 190°F collector outlet), an electric motor is provided to operate the water chiller independently or to make up the difference for the required horsepower and that supplied by the Rankine engine. Storage is used to supply energy in the same manner as in the heating subsystem, except pump P_1 is the prime mover in the glycol/water loop. The baseline design uses a constant-speed compressor input and therefore the electric motor is on-line at all times, supplying the balance of the required horsepower.

A variation of this concept is under study which uses a variable-speed R/C input to the compressor and thus variable cooling output. The auxiliary motor operates only when the water chiller controls indicate the cooling load is not satisfied. In this mode the water chiller operates as a constant-speed system delivering a maximum of 25 tons of cooling.

4.14.3.3 Specific Subsystem/Component Design Criteria -- The heating/storage circuit pump, P_2, and the cooling circuit pump, P_1, are located ahead of the loads so as to provide a common location for the expansion tank. This provides for air elimination and system pressure control during all modes of operation.

When the system is heating or cooling from storage, control valve V_2 diverts flow around the solar collectors.

When the system is in the purge mode, control valve V_3 diverts flow through the purge coil unit which is placed downstream of the solar collectors for protection of other system components from overtemperature.

In the water storage loop, the storage pumps P_3 and P_4 are located so as to provide a counter-flow arrangement at the storage heat exchanger. Two pumps are used to take advantage of stratification of storage temperature within the tank. The common return line from the storage heat exchanger enters the middle of the storage tank. Pump P_3 draws hotter water from the top of the storage tank for heating or cooling from storage modes of operation. Pump P_4 draws cooler water from the bottom of the storage tank for storage charging of collected solar energy.

The solar domestic hot-water preheater, in combination with the storage heat exchanger, provides a double-wall separation between the ethylene glycol/water solution in the collector loop and the potable domestic hot-water system.

The solar domestic hot-water preheater heats incoming cold water with solar-heated storage water through a tube bundle circulated by pump P_6. Storage water is drawn from the top of the storage tank and returned to the bottom to take advantage of thermal stratification.

A thermostatic mixing valve keeps the temperature below 140°F. A conventional water heater is available downstream to add auxiliary energy as required to obtain 140°F water.

The location of each of the components within the system is determined by the following criteria:

- Optimize performance of entire system.
- Provide reliable operation of each component.
- Provide control functions for each mode of operation.

The storage heat exchanger, within the collector loop, is placed downstream of the solar collectors and ahead of the loads so as to provide a means for inlet temperature control to the loads. This also provides for a convenient method for simultaneous storage charging.

The heating/storage circuit and the cooling circuit are placed in parallel, downstream of the storage heat exchanger. Each circuit is controlled by its pump and protected from backflow by a check valve.

4.14.4 Commercial Building Heating and Cooling System Description

The proposed system for a commercial heating and cooling system is a single-loop, solar-powered, two-pipe hydronic heating and cooling system with a separate water storage and domestic hot-water heating loop. Being a two-pipe design, the central system is either in the heating mode or the cooling mode as determined by a Cooling Load Analyzer. The central heating is provided by either direct or stored solar energy and the central

cooling is provided by three solar-powered Rankine engine/auxiliary electric motor-driven water chillers. The system is composed of the following major components:

- Liquid-cooled flat-plate collectors
- A water storage tank
- A solar-fired domestic water preheater with storage water tube bundle and circulating pump
- A gas-fired hot-water heater
- Water chillers driven by a Rankine engine and/or an auxiliary electric motor and controlled by a Cooling Load Sequencer
- A tube-and-shell heat exchanger, four central system pumps and associated piping and valving
- A condenser water pump and cooling tower
- A control system
- An air-cooled heat purge unit with fan
- Four individual subsystems consisting of a circulating pump and a gas-fired air-handling unit with a water heating/cooling coil

The arrangements within the system is as shown in Figure 4-56. It consists of a glycol collector loop which interfaces with a water storage loop, through a tube-and-shell heat exchanger.

The glycol collector loop consists of the solar collectors, the shell side of the storage heat exchanger, the heating coil and pump P_2, the Rankine boilers and pump P_1, the purge coil, and three control valves as required for the different modes of operation.

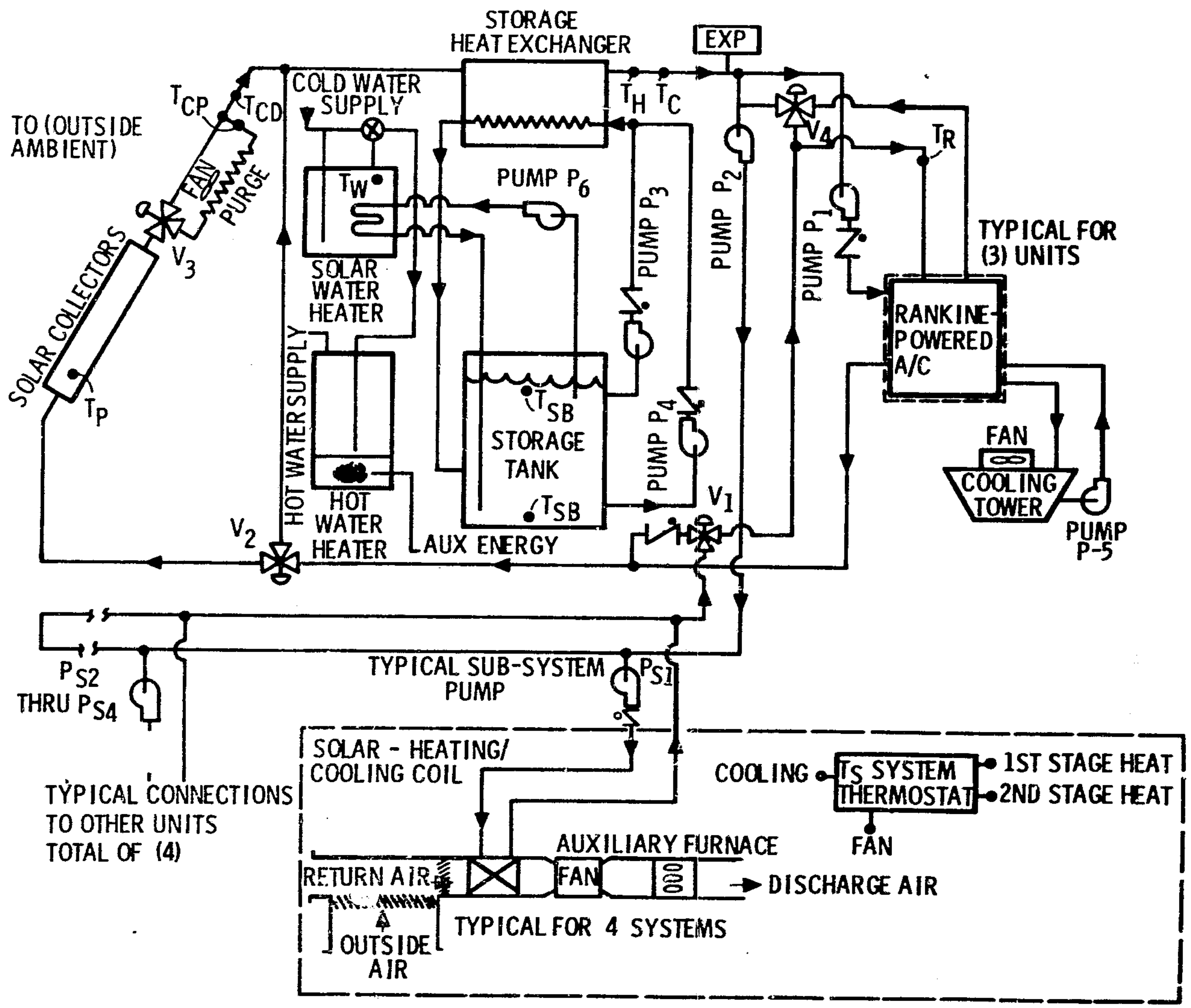

Figure 4-56. Commercial Building H/C System

The water storage loop consists of the storage tank, storage pumps P_3 and P_4, and the tube side of the storage heat exchanger, and the solar DHW preheater pump, P_6.

The system provides 10 modes of operation:

- Direct heating from collectors
- Direct heating from storage
- Direct heating and storage simultaneously
- Auxiliary heating (insufficient solar)
- Rankine cooling from collectors
- Rankine cooling from storage
- Rankine cooling and storage simultaneously
- Electric motor auxiliary cooling
- Domestic hot-water preheater
- Purge excess energy

4.14.4.1 Heating Subsystem Operation -- When solar energy is available and heating is required, the collectors supply heat directly to the system. Pump P_1 provides the heat-transfer fluid movement in the main loop, the subsystem pump circulates the subsystem loop, and the air-handling unit moves the building air over the heat coil.

During periods of high solar radiation and low heating demand, both heating and storage loop operate simultaneously, with the storage charge pump, charging the storage tank by removing water from the bottom, adding energy in the heat exchanger and returning it to the center of the storage tank, thus taking advantage of thermal stratification. During periods of high solar radiation and low heating demand with full storage charge, the system

temperatures will increase and as an overtemperature protective device, the purge coil operates, controlling the downstream temperatures to a preselected value.

When solar energy is not available and heating is required, storage supplies heat to the furnace through the heat exchanger. Pump P_2 drives the main loop and pump P_3 extracts heat from the top of the storage tank and returns it to the center, again taking advantage of the tank stratification. If the storage tank temperature is not high enough to supply heating, the heating load cannot be satisfied by the solar system and the second-stage thermostat activates the auxiliary gas burner until a comfortable temperature is maintained.

4.14.4.2 Cooling Subsystem Operation -- When solar energy is available and cooling is required, the collectors supply heat directly to the Rankine boiler. Pump P_1 provides the heat-transfer fluid movement and, if necessary, can be sized differently than P_2 to improve efficiency of the collector and Rankine-engine operation. Each Rankine engine drives a high COP water chiller, which, in turn, provides chilled water to the system. When the cooling demand is satisfied, pump P_2 is shut down and the system reverts to the storage mode explained in the heating subsystem. During high solar radiation and low cooling demand, simultaneous cooling and storage is available. The purge coil unit will protect the system from excessive temperatures.

When solar energy is not available or insufficient to operate the Rankine engine at the design horsepower (20 horsepower at 190°F collector outlet), an electric motor is provided to operate the water chiller independently or to make up the difference for the required horsepower and that supplied by the Rankine engine. Storage is used to supply energy in the same manner as in the heating subsystem, except pump P_1 is the prime mover in the glycol loop. The baseline design uses a constant-speed compressor input and therefore the electric motor is on-line at all times, supplying the balance of the required horsepower.

A variation of this concept is under study which uses a variable-speed R/C input to the compressor and thus variable cooling output. The auxiliary motor operates only when the water chiller controls indicate the cooling load is not satisfied. In this mode each water chiller operates as a constant-speed system delivering a maximum of 25 tons of cooling.

4.14.4.3 Specific Subsystem/Component Design Criteria -- The heating/storage circuit pump, P_2, and the cooling circuit pump, P_1, are located ahead of the loads so as to provide a common location for the expansion tank. This provides for air elimination and system pressure control during all modes of operation.

When the system is heating or cooling from storage, control valve V_2 diverts flow around the solar collectors.

When the system is in the purge mode, control valve V_3 diverts flow through the purge coil unit which is placed downstream of the solar collectors for protection of other system components from overtemperature.

In the water storage loop, the storage pumps, P_3 and P_4, are located so as to provide a counter-flow arrangement at the storage heat exchanger. Two pumps are used to take advantage of stratification of storage temperature within the tank. The common return line from the storage heat exchanger enters the middle of the storage tank. Pump P_3 draws hotter water from the top of the storage tank for heating or cooling from storage modes of operation. Pump P_4 draws cooler water from the bottom of the storage tank for storage charging of collected solar energy.

The solar domestic hot-water preheater, in combination with the storage heat exchanger, provides a double-wall separation between the ethylene glycol/water solution in the collector loop and the potable domestic hot-water system.

The solar domestic hot-water preheater heats incoming cold water with solar heated storage water through a tube bundle circulated by pump P_6. Storage water is drawn from the top of the storage tank and returned to the bottom to take advantage of thermal stratification.

A thermostatic mixing valve keeps the temperature below 140°F. A conventional water heater is available downstream to add auxiliary energy as required to obtain 140°F water.

The location of each of the components within the system is determined by the following criteria:

- Optimize performance of entire system.
- Provide reliable operation of each component.
- Provide control functions for each mode of operation.

The storage heat exchanger, within the collector loop, is placed downstream of the solar collectors and ahead of the loads so as to provide a means for inlet temperature control to the loads. This also provides for a convenient method for simultaneous storage charging.

The heating/storage circuit and the cooling circuit are placed in parallel, downstream of the storage heat exchanger. Each circuit is controlled by its pump and protected from backflow by a check valve.

4.14.5 Solar HVAC System Design Tradeoffs

Performance simulations and economic analyses of the selected solar-assisted heating and cooling systems were performed for systems for single-family residences, multifamily residences and commercial buildings. The tradeoff studies began with a "baseline" system, and various parameters that are significant to both cost and performance were varied separately to determine their individual effects on system performance and economics. The effects of these parameter variations have been studied for Atlanta, Georgia. The magnitude of these effects would be different in other localities.

4.14.5.1 Single-Family Residence (SFR) -- The baseline solar system modeled in the digital simulation program is shown schematically in Figure 4-54. It consists of a collector array with piping headers on both sides of the collectors for inlet and outlet of the collector fluid, a collector with two glass covers and a single storage tank filled with water for sensible heat storage. The system features direct flow of collector fluid to the Rankine cycle boiler or the house heating coil.

The most important system variable which greatly affects both performance and economics is the collector area. The collector area for the SFR was varied from 480 ft^2 to 1260 ft^2. Figure 4-57 shows the percent of solar energy supplied to the Atlanta SFR heating, cooling and hot water loads. The solar contribution of these loads varies from about 20 percent, 42 percent and 77 percent for 480 ft^2 of collector area to about 72 percent, 75 percent and 90 percent for 1260 ft^2 system, respectively. The baseline 756 ft^2 collector area supplies 60 percent, 47 percent and 85 percent of the home's heating, cooling and hot water loads, respectively. These predictions were made for a system with a 1000-gallon storage tank.

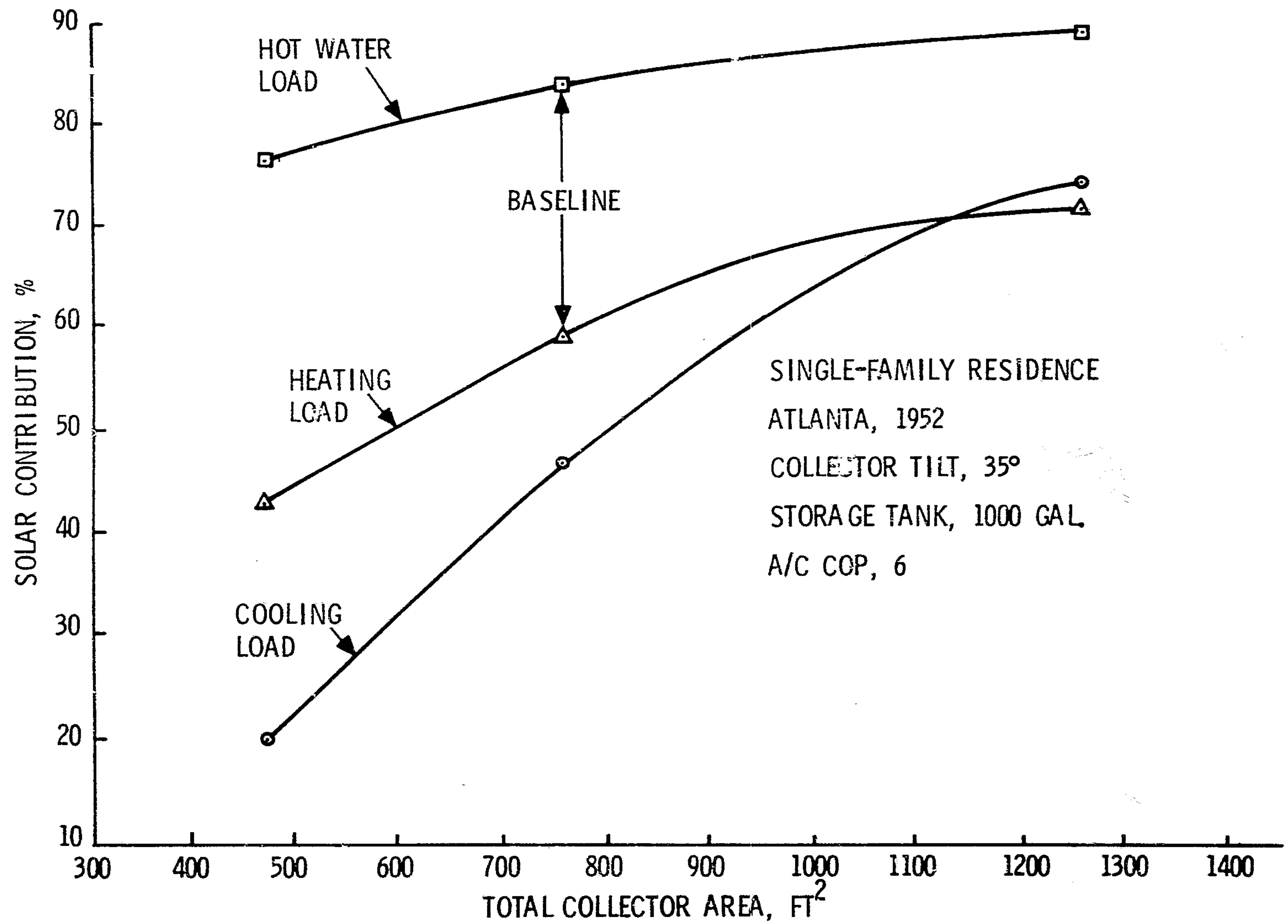

Figure 4-57. Solar Contribution versus Collector Area - SFR

It is obvious that the percent of energy supplied increases as the system collector area increases. However, at approximately 1100 ft^2, a break in the rate of relative contribution occurs. The solar contribution at this point to both heating and cooling is 70 percent. Figure 4-58 shows the additional auxiliary electric power required to support the solar cooling subsystem for the various collector areas.

Figure 4-58 also presents the impact of R/C design point thermodynamic efficiency on auxiliary cooling power. The baseline Rankine cycle design features a shaft output of 2.4 hp at an efficiency of 8 percent at 190°F. A decrease in this cycle efficiency increases the thermal energy input requirements to the R/C boiler. Figure 4-59 presents the impact of R/C efficiency on a solar contribution to the SFR cooling load. The auxiliary electric cooling power logically increases as η_{RC} decreases; however, an equally significant result is that the solar contribution decreases rapidly.

Both Figures 4-58 and 4-59 may be used to determine whether it's more cost effective to increase overall cooling system performance through improvements in R/C parameters (heat exchanger effectiveness, regenerators, flow rate, etc.) or by merely adding collector area. For example, Figure 4-60 indicates that η_{RC} may be improved from 6.5 percent to near 8 percent through the addition of a regenerator and an increase in heat exchanger effectiveness. However, this would cost about $600.00. To achieve the same cooling system performance (i.e., auxiliary power consumption) for an η_{RC} = 6.5 percent. Figure 4-58 indicates that an additional 170 ft^2 of collectors would be required. However, these installed collectors would cost approximately 170 ft^2 x \$17.85/ft^2 = \$3,034.00 and thus it may be concluded that improvements in R/C parameters for cooling performance is more cost effective than adding collection area.

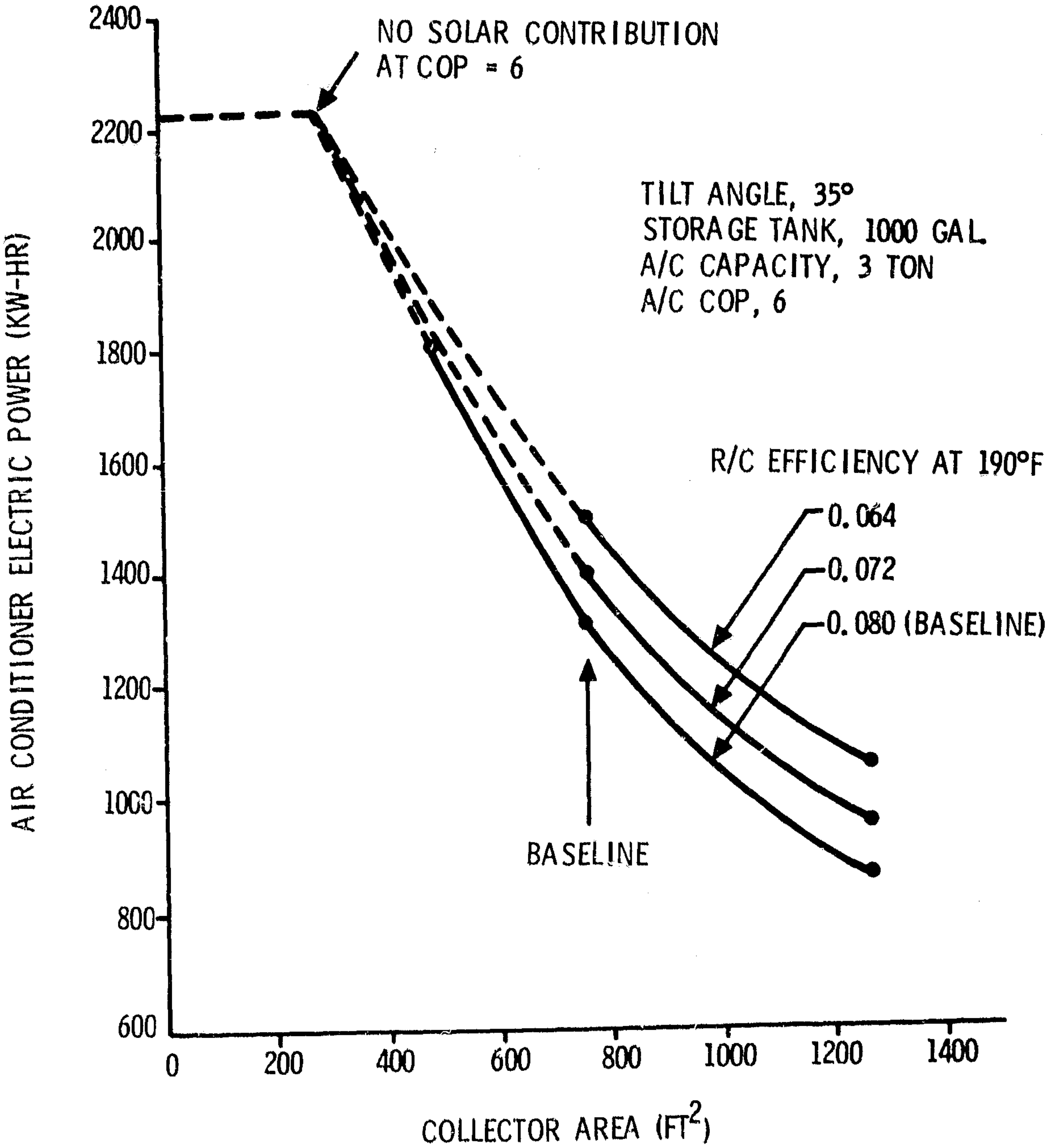

Figure 4-58. Cooling Auxiliary Power versus Collector Area - SFR

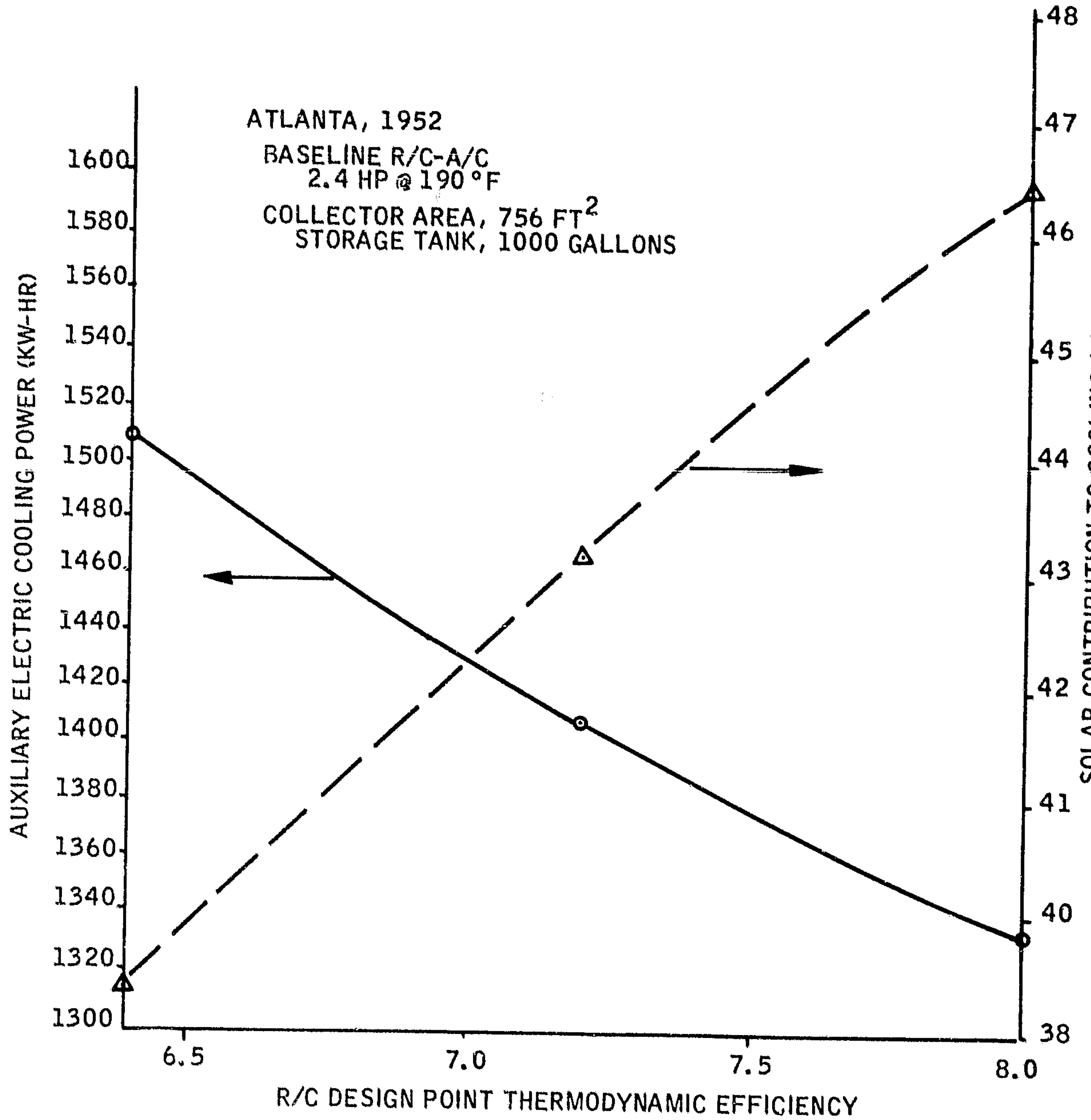

Figure 4-59. Cooling Subsystem Performance versus R/C Efficiency - SFR

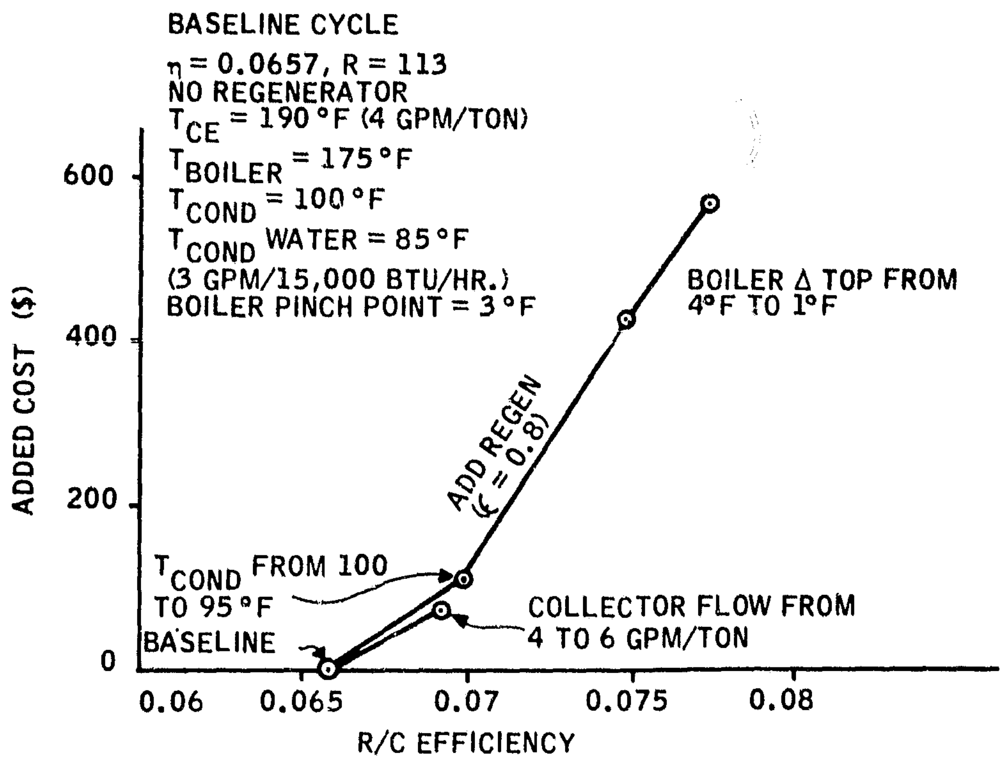

Figure 4-60. 3-Ton Cycle Improvement Costs

The impact of the vapor compression air conditioner COP on auxiliary cooling power and solar contribution is shown in Figure 4-61. The baseline COP for the A/C under constant speed control is 6. These results indicate that COP is one of the most important parameters for cooling system performance.

The baseline design would require 1300 kW-hr of electricity while a conventional A/C system with an energy efficiency ratio (EER) of 6.1 would require 5926 kW-hr for the year's cooling requirements.

The importance of collector tilt angle on the contributions of solar energy to the SFR heating, cooling and hot water loads is shown in Figure 4-62. It is obvious that tilt angle is not as sensitive a parameter as collector area. Contributions to heating and hot water loads do not vary significantly for a tilt angle variation from 20 to 50 degrees. Cooling contribution appears near maximum at 20 degrees tilt. It must be repeated that tilt angle and relative solar contributions are site specific and that a different mix of relative solar contributions and sensitivity to tilt angle can be expected in different locations and different latitudes.

The baseline tilt angle for this study is 35 degrees (the approximate latitude of Atlanta). The collector tilt angle of 20 degrees is approximately the same angle as typical roof construction (4' in 12' or 18.4 deg) and thus from an architectural as well as optimized cooling contribution, a tilt angle of 20 degrees may be a logical choice.

Figure 4-63 shows the number of solar Btus supplied directly to the SFR heating, cooling and hot water loads and the total Btus collected for the year. At 35 degrees tilt, both the number of Btus collected and supplied directly to the loads is maximum. The energy difference between the Btus collected and those supplied directly to the loads is composed of the following:

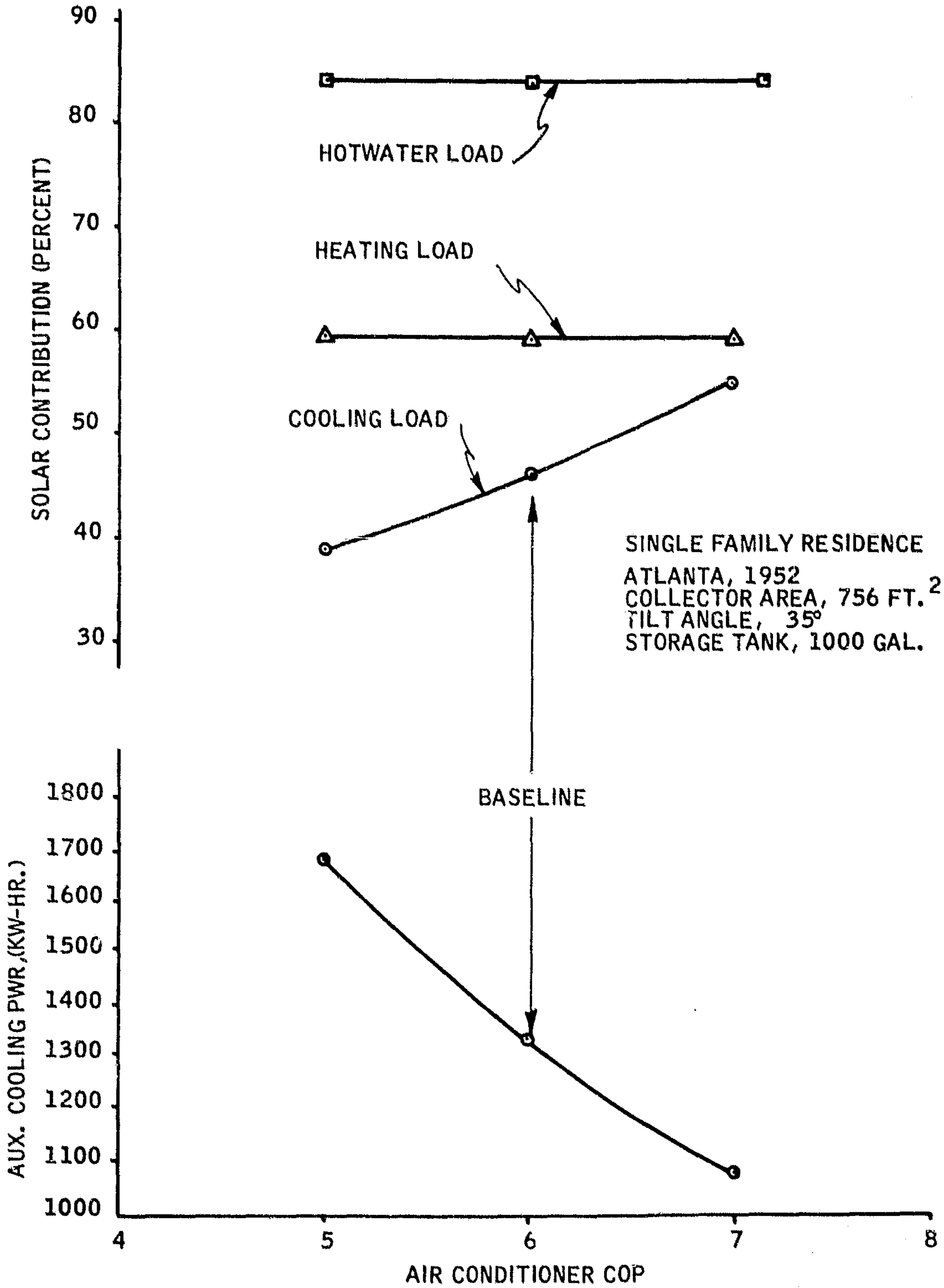

Figure 4-61. Cooling Subsystem Performance versus Air Conditioner COP - SFR

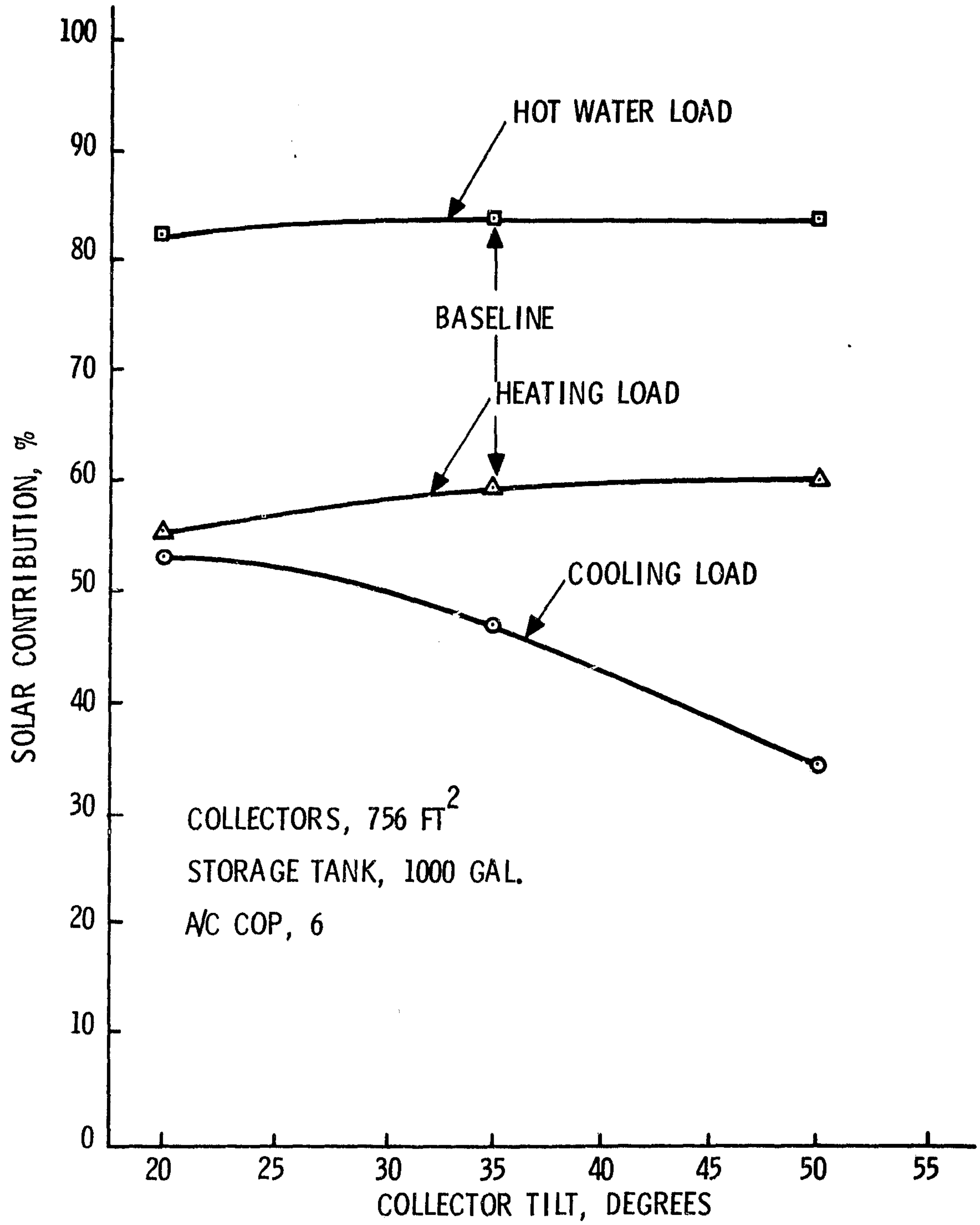

Figure 4-62. Solar Contribution versus Collector Tilt - SFR

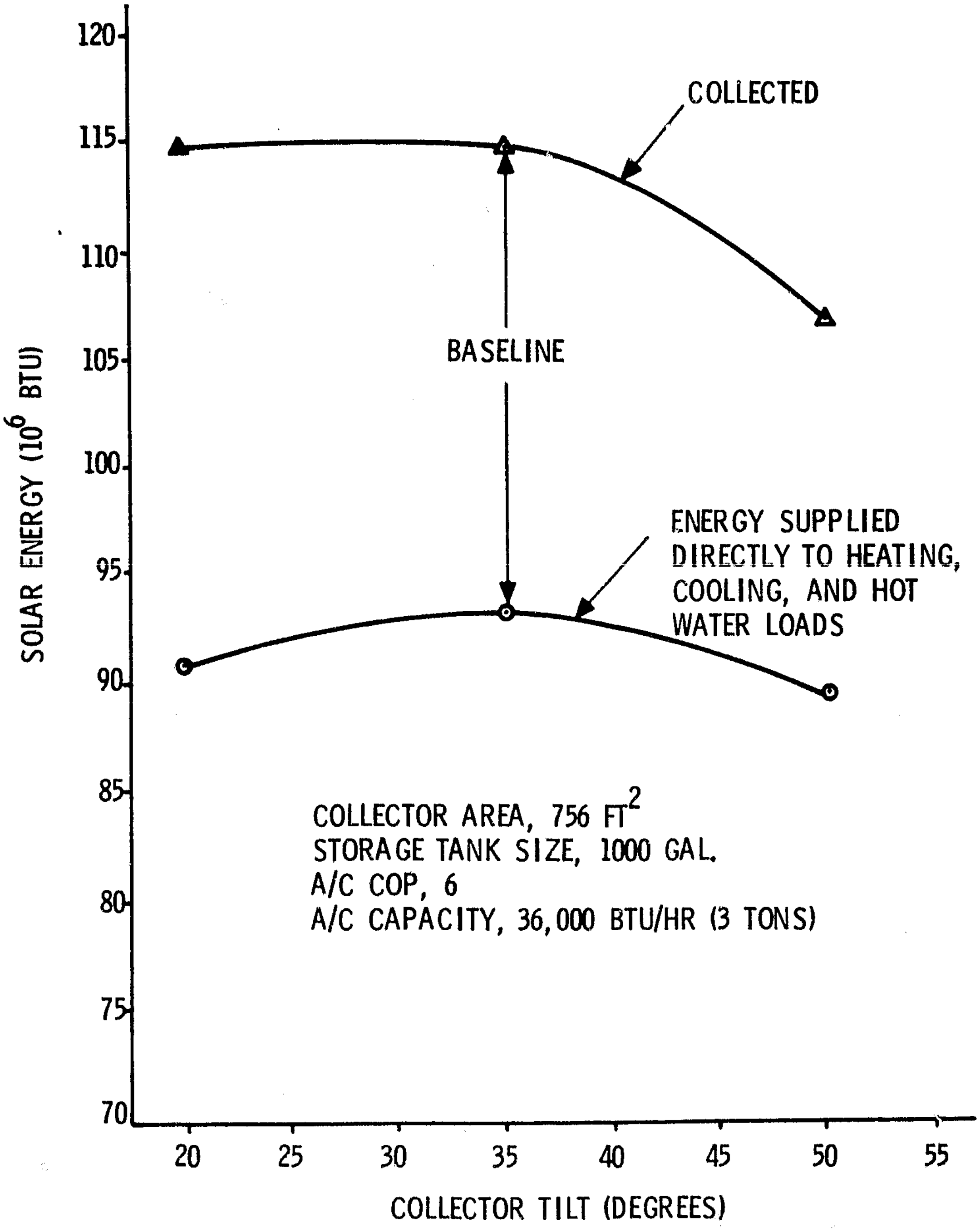

Figure 4-63. Solar Energy Supplied versus Collector Tilt - SFR

a) The efficiency of the Rankine cycle is less than 10 percent. In defining Btus supplied to the cooling load what is meant is solar Btus that actually are converted to useful shaft power and thus useful cooling capacity. The efficiency (or accurately, inefficiency) of the R/C results in most of the Btus that are consumed by the R/C being rejected out to the cooling tower. This element, which cannot be considered a system loss in the same light as storage tank heat loss, makes up the major difference between Btus collected and Btus supplied directly to the SFR loads, as shown in Figure 4-63.

b) A small portion of the difference is related to the purge mode. When the collector loop temperature exceeds 230°F, the heat purge rejection system is activated and the loop temperatures are maintained below a specified set point (i. e., 220°F). Thus, in the purge mode (only 106 hours for the Atlanta year, 1952) solar Btus were counted as being collected but they were also rejected immediately. Only a very small portion of the difference shown in Figure 4-63 is attributable to these purge conditions.

c) Storage tank heat losses also represent a very small portion of the difference shown in Figure 4-63.

The baseline tilt angle of 35 degrees has been selected based on the maximum solar Btus supplied directly to the SFR heating, cooling and hot water loads.

The performance of the system with the collector array facing away from south was not simulated. It is well known that the optimum system performance is achieved with collectors facing due south. Previous calculations have shown that variations of 30 deg east or west reduce performance about 3 percent.

The performance of a 756 ft^2 solar HVAC system with different size storage tanks is shown in Figure 4-64. The percent of solar energy supplied to the heating load varies from about 52 percent for a 500-gallon tank to approximately 62 percent for a 1500-gallon tank. A 1000-gallon storage tank was selected as the baseline size based on the reasonable mix of solar contributions shown in Figure 4-64 and on the near minimum in auxiliary electric power required for the SFR cooling load.

Heat losses from the solar HVAC storage tank were neglected for these system tradeoff analyses. However, storage tank heat losses were included in the baseline system performance analysis discussed subsequently in subsection 4.14.6. Analysis performed previously showed that a tank UA of 9.44 (4-inch thick of fiberglass insulation), the solar heating system would provide about 2 percent less energy over the year than if heat losses were completely eliminated. If the storage tank is located inside the home, the heat loss would decrease the home's heat load and the net effect would be the same as a perfectly insulated storage tank. Of course, storage tank heat loss would increase the home's cooling load if the tank were located inside the home. Thus, tank losses were neglected for these tradeoff analyses so as to not influence the tradeoff conclusions.

The baseline solar HVAC system for the SFR has a heat exchanger (Hx) between the solar collector glycol loop and the hot water storage tank subsystem. The effectiveness of this heat exchanger on system performance is shown in Figure 4-65. The baseline Hx effectiveness was assumed to be 0.55, a reasonable level of performance for a glycol/water exchange condition for the flow rates defined for the baseline solar system. Figure 4-65 indicates that solar contributions to the house heating, cooling and hot water loads are not strongly influenced by this heat exchanger's effectiveness. Note that this result may offer an area for system cost reduction since annual electric power for cooling is only affected by about 3 percent.

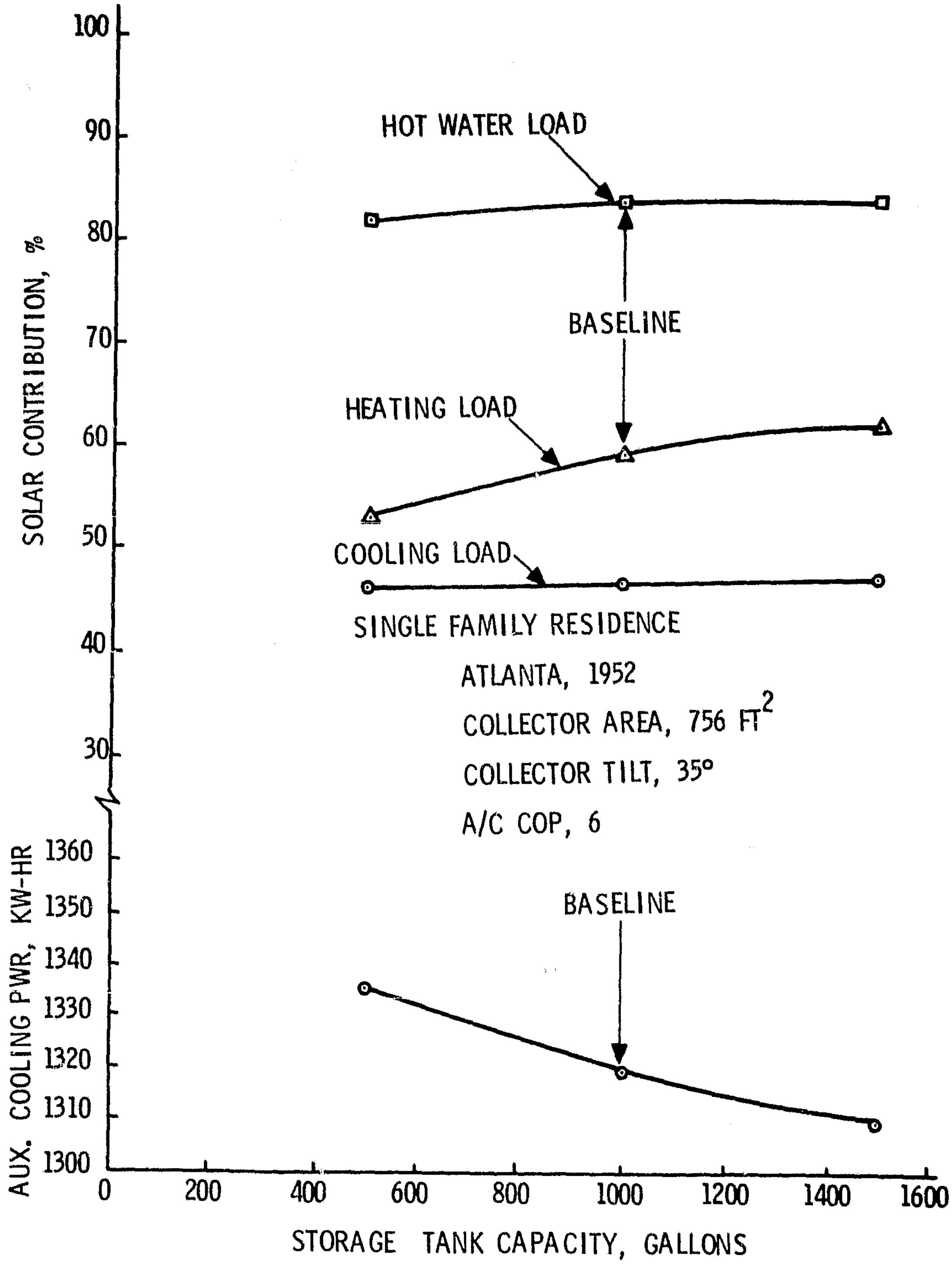

Figure 4-64. Subsystem Performance versus Storage Tank Capacity - SFR

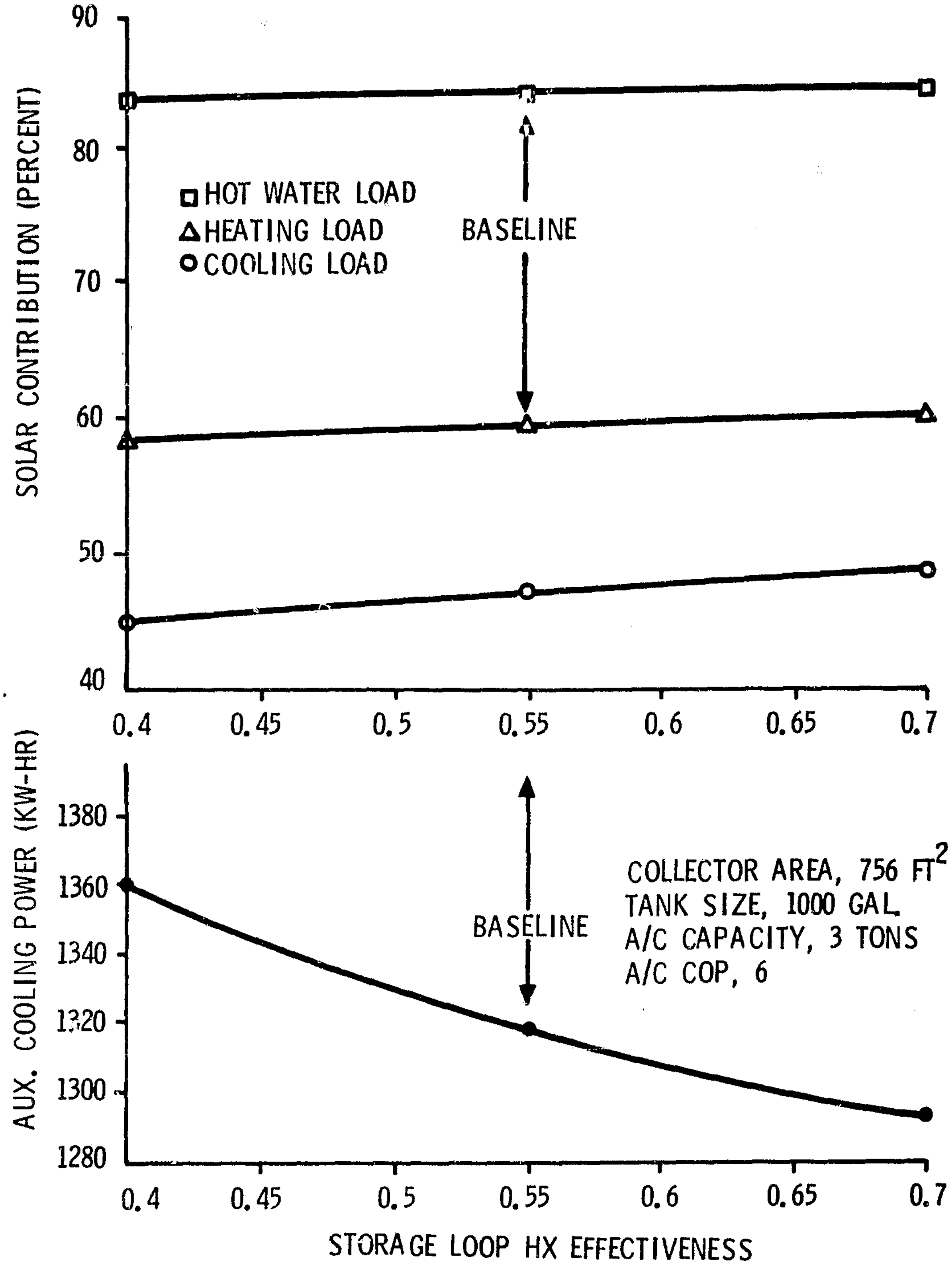

Figure 4-65. Subsystem Performance versus Heat Exchanger Effectiveness - SFR

A method to cut the cost of solar energy systems is to reduce the piping and solar collector costs. This can be partially accomplished by mounting two flat-plate collectors in series. The exit flow of one collector enters the second collector directly. Fluid exiting from the second collector then goes into the piping header. The only disadvantage of this arrangement is that the performance of the second collector in series is slightly degraded because the fluid inlet temperature is higher. The system performance of a solar HVAC system with two collectors arranged in series has been shown in the past to be only degraded by a fraction of a percent.

In conclusion, a system analyses has been performed to determine the effect of variations in subsystem parameters on overall solar HVAC system performance. A baseline system for a single-family residence was established from these tradeoff studies and the thermal performance and economic analysis of this system are discussed subsequently in subsection 4.16.6.1 This baseline system features the following parameters for the Atlanta, Georgia SFR.

• Collector tilt angle	35 deg
• Number of collectors	42
• Total collector area	756 ft^2
• Storage tank capacity	1000 gallons
• Storage tank heat exchanger effectiveness	0.55
• Air conditioner capacity	3 tons
• Air conditioner COP	6
• Heating system capacity	60,000 Btu/hr

4.14.5.2 <u>Multifamily Residence (MFR)</u> -- The baseline solar system simulated by the computer program was discussed earlier in this section and is shown in Figure 4-52. The system has the same important features as the Atlanta SFR System. The results of the SFR system tradeoff analyses

indicated that a collector tilt angle of 35 deg was an optimum selection. Thus, the two major system variables that were analyzed for the MFR were collector area and storage tank size.

Figure 4-66 presents the effect of collector area variation on solar HVAC performance. At the baseline area of 6300 ft^2, the solar HVAC supplies 42 percent, 65 percent, and 84 percent of the building's cooling, heating and hot water loads, respectively. Increasing the area to 10,800 ft^2, can raise the solar contribution to the same respective loads to 67.5 percent, 77 percent and 89.5 percent. The auxiliary electric power required for the water chiller subsystem is shown in Figure 4-67. Increasing the collector area from the baseline of 6300 ft^2 to 10,800 ft^2 will result in a reduction of the annual cooling electric power from 12,700 kW-hr to 8600 kW-hr.

The effect of storage tank capacity on system performance is shown in Figure 4-68. Again, as in the SFR system, storage tank size does not strongly influence overall system performance parameters.

The baseline MFR annual thermal performance and economic analysis is discussed subsequently in subsection 4.16.6.2. This baseline features the following system parameters for the Atlanta MFR.

• Collector tilt angle	35 deg
• Number of collectors	350
• Total collector area	6300 ft^2
• Storage tank capacity	8333 gallons
• Storage tank heat exchanger effectiveness	0.55
• Water chiller capacity	25 tons
• Chiller COP	6
• Heating system capacity	600,000 Btu/hr

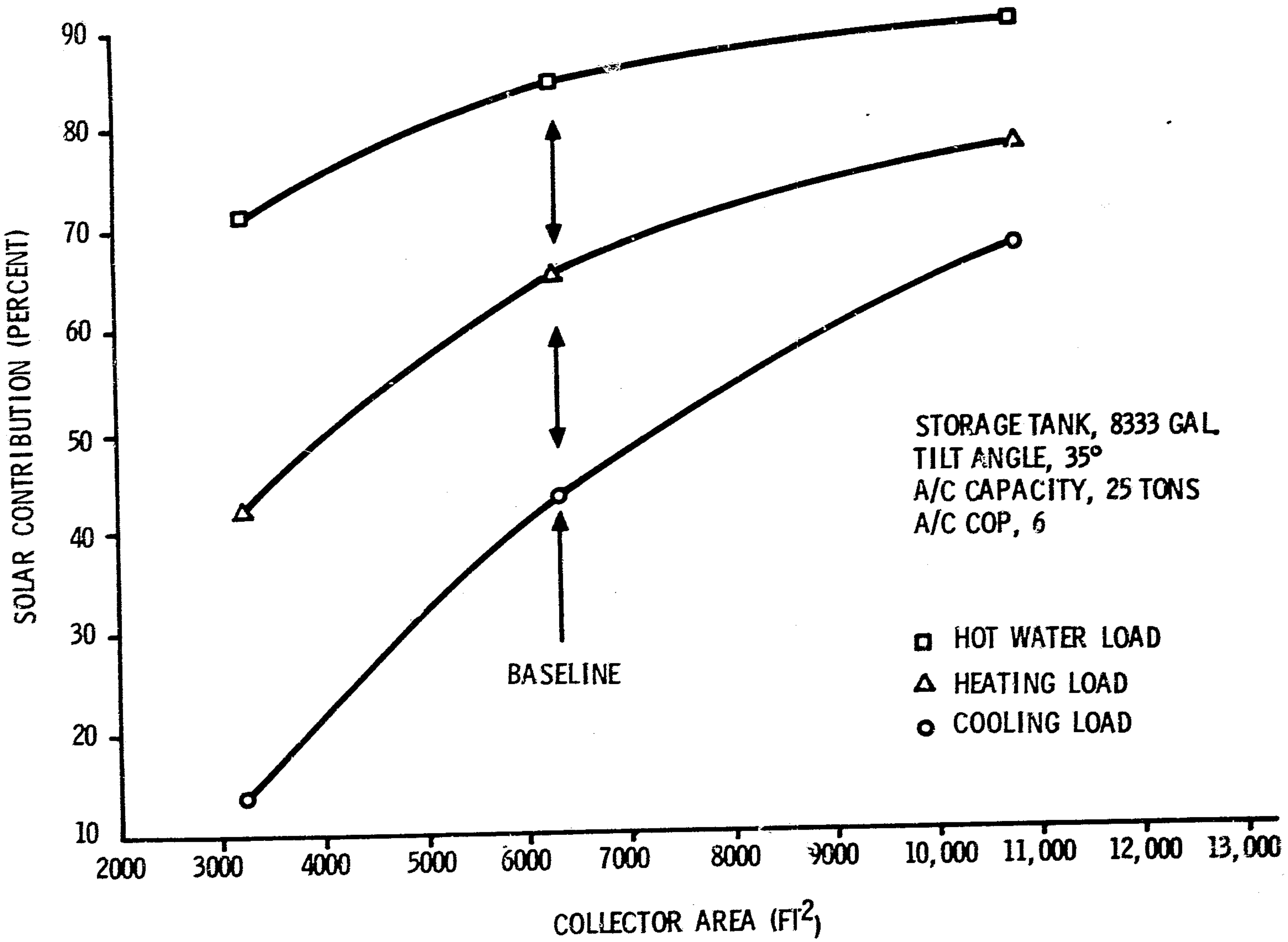

Figure 4-66. Solar Contribution versus Collector Area - MFR

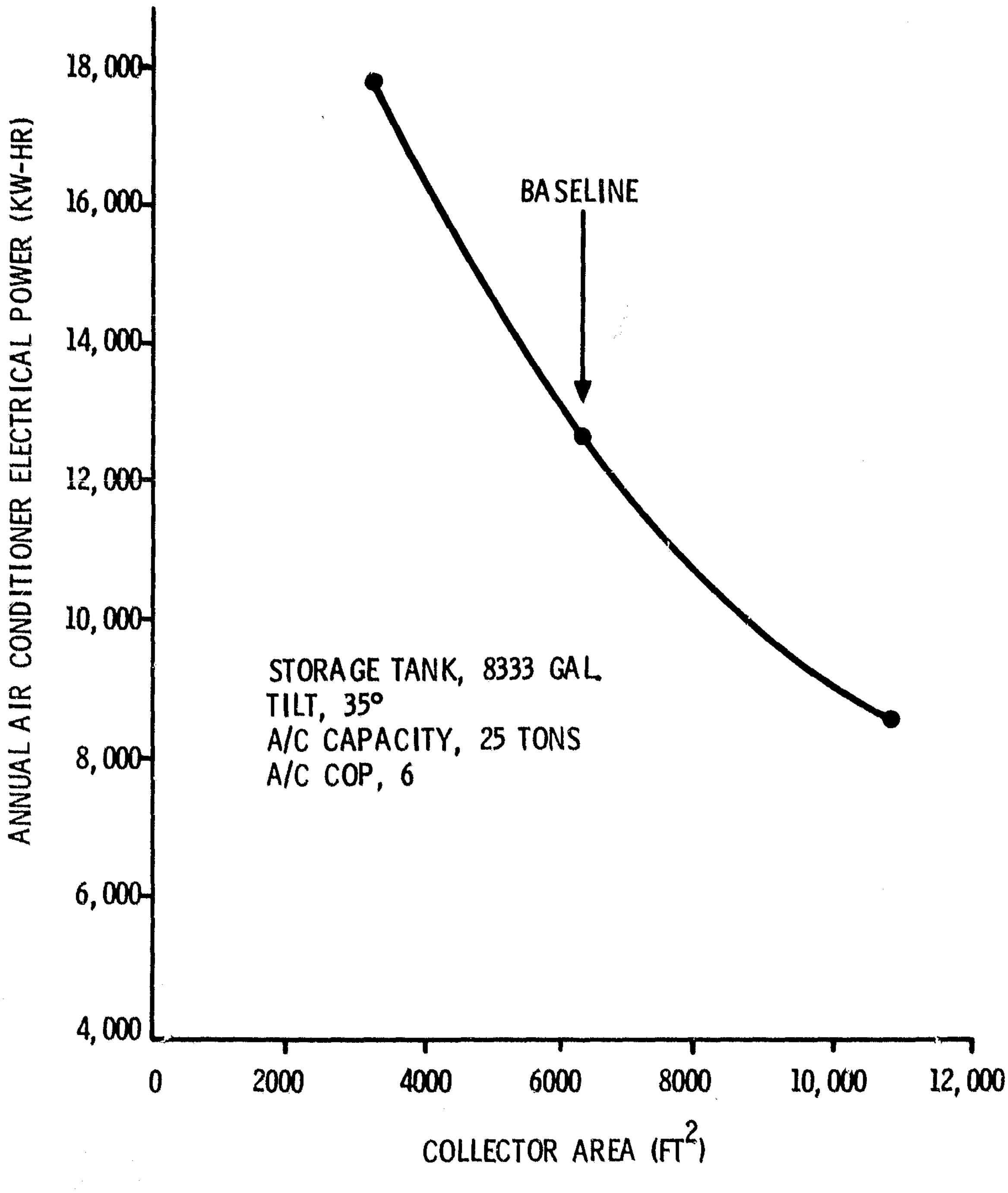

Figure 4-67. Cooling Auxiliary Power versus Collector Area - MFR

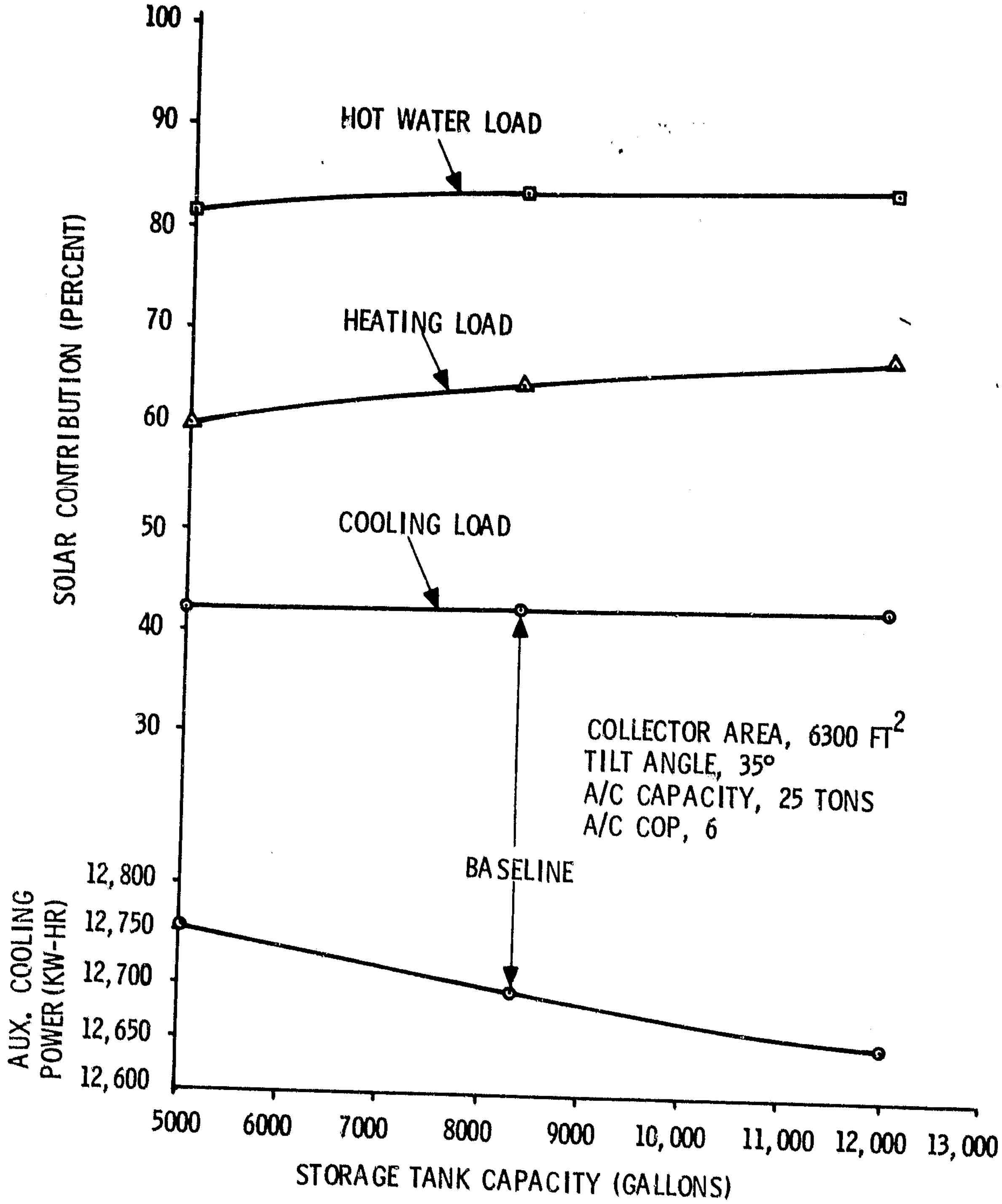

Figure 4-68. Subsystem Performance versus Storage Tank Capacity - MFR

4.14.5.3 Commercial Building -- A baseline solar HVAC system was formulated for the commercial building model in Atlanta, Georgia, based on three 25-ton R/C-A/C subsystems performing the cooling task. The thermal performance of this large solar HVAC system is discussed in subsection 4.16.6.3. This system, which is merely an expansion of the MFR solar system, has the following system features:

• Collector tilt angle	35 deg
• Number of collectors	670
• Total collector area	12,060 ft^2
• Storage tank capacity	15,600 gallons
• Water chiller capacity (three 25-ton R/C-A/C units)	75 tons
• Water chiller COP	6
• Heating system capacity	800,000 Btu/hr

4.14.6 Baseline System Performance and Economic Analysis

4.14.6.1 Single-Family Residence System -- The recommended solar assisted heating, cooling and hot water system for a single-family residence is a hydronic-to-warm air heating system, a Rankine cycle-vapor compression air conditioner operated at constant speed, and a domestic hot water preheat system. The baseline system consists of the major components shown in Figure 4-51 and listed in Table 4-6.

The performance of this solar HVAC system for an Atlanta, Georgia, house (2010 ft^2) is shown in Figures 4-69 through 4-71. The space heating load

Table 4-6. Single-Family Residence (2010 ft^2) Atlanta, Georgia

Baseline Solar HVAC System Design:

a) Cooling Subsystem

3-ton air conditioner
COP = 6
Constant speed control
Rankine cycle design point
2.4 hp at 190°F
$\eta_{R/C}$ at 190°F = 8 percent
Control range, 150°F - 200°F

b) Heating Subsystem

Capacity, 60,000 Btu/hr
Hx coil effectiveness = 0.6

c) Storage Subsystem

Tank capacity, 1000 gallons
Storage Hx effectiveness, 0.55

d) Solar Collector Subsystem

Area 756 ft^2 (total) 625 ft^2 (effective)
Number of collectors, 42
Flow rate, 12 gpm

e) House Loads

UA = 894 Btu/hr°F
Infiltration, 268 cfm (day) 220 cfm (night)
Internal load, 2550 Btu/hr (day) 1350 Btu/hr (night)
Sun load on 80 ft^2 windows
Design day heating load, 51,835 Btu/hr based on
T_{AM} = 23°F, T_{House} = 70°F
Design day cooling load, 33,971 Btu/hr (2.83 tons) based on
T_{AM} = 92°F, T_{House} = 76°F

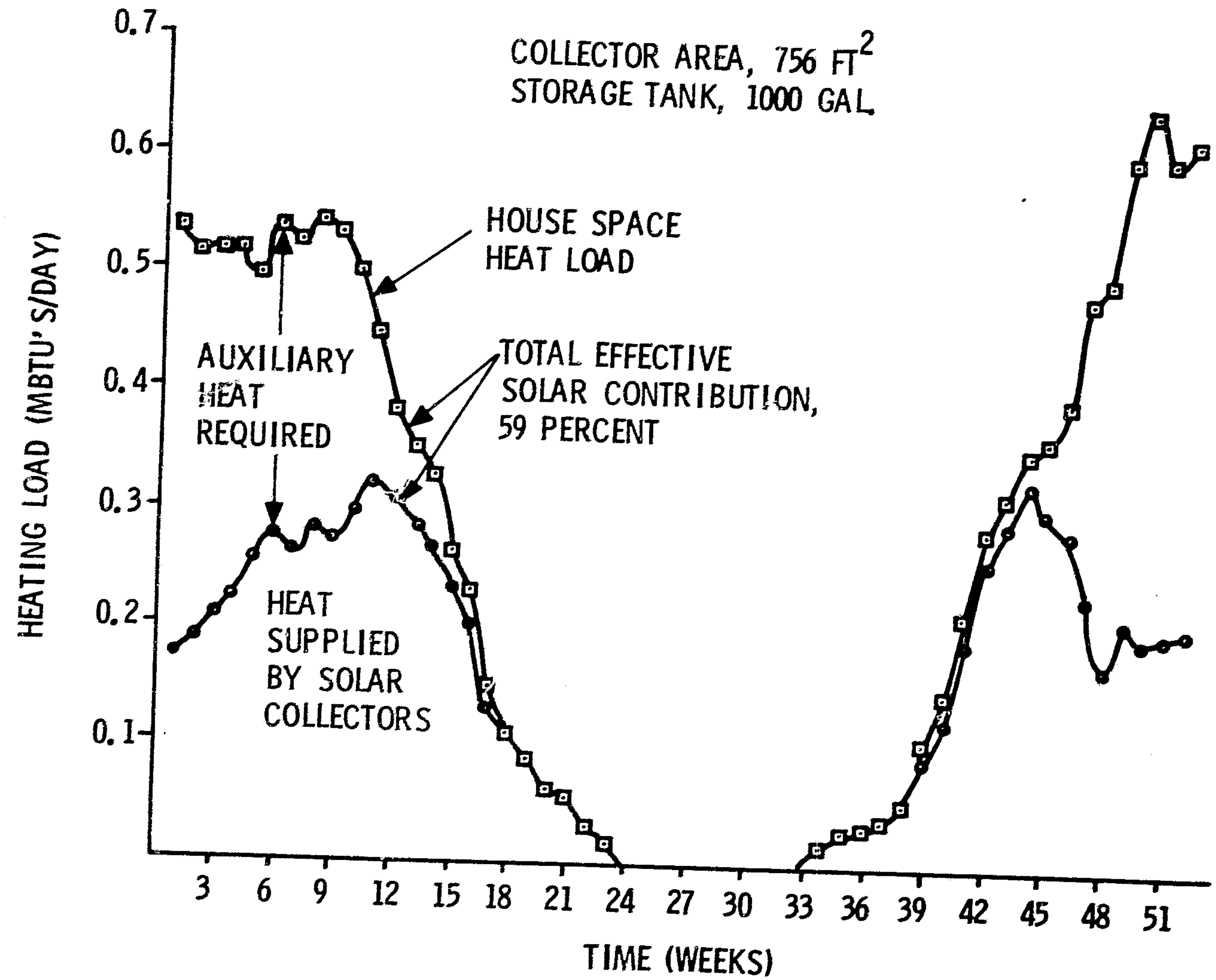

Figure 4-69. Weekly Solar Contribution to Heating Load - SFR

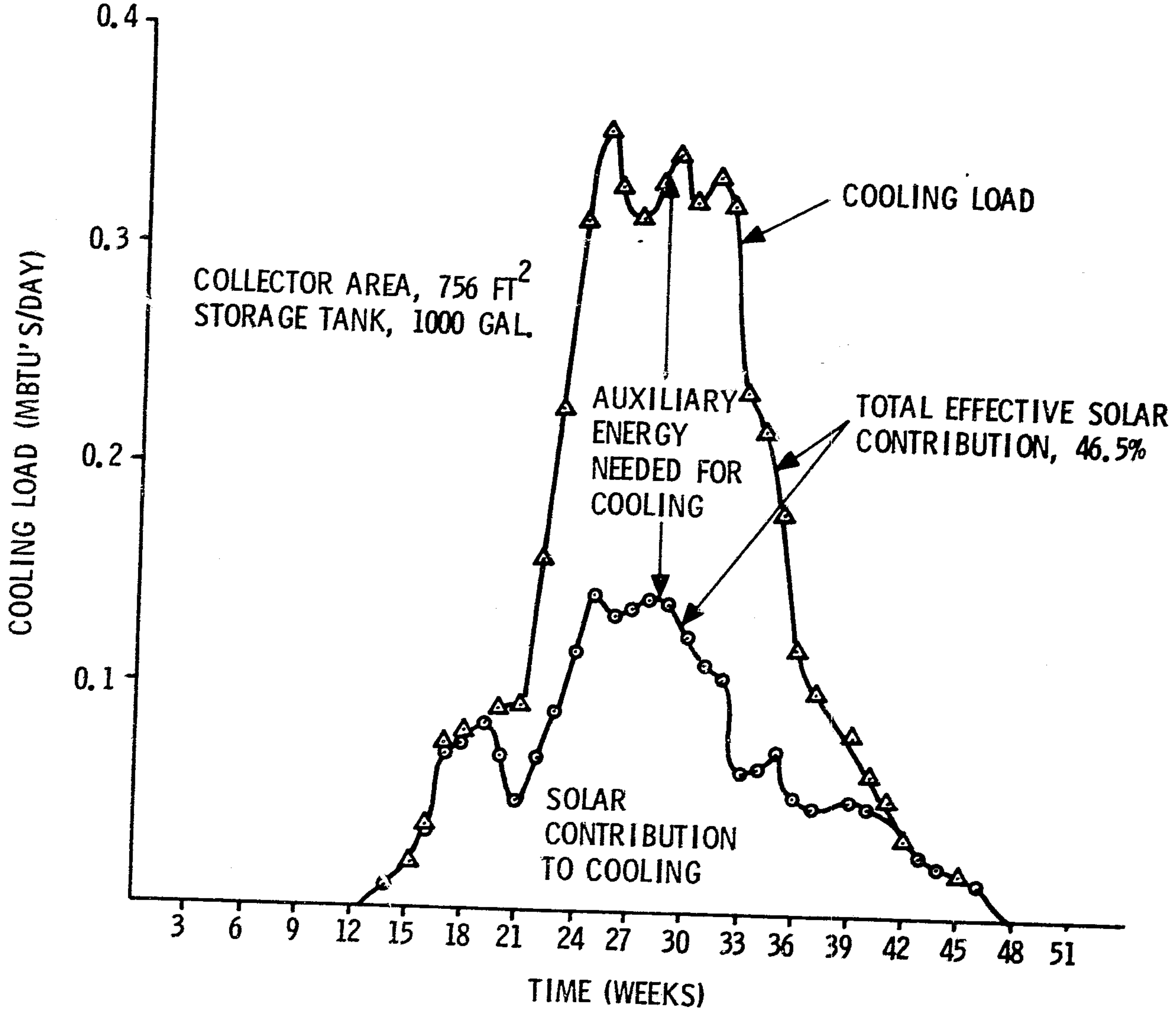

Figure 4-70. Weekly Solar Contribution to Cooling Load - SFR

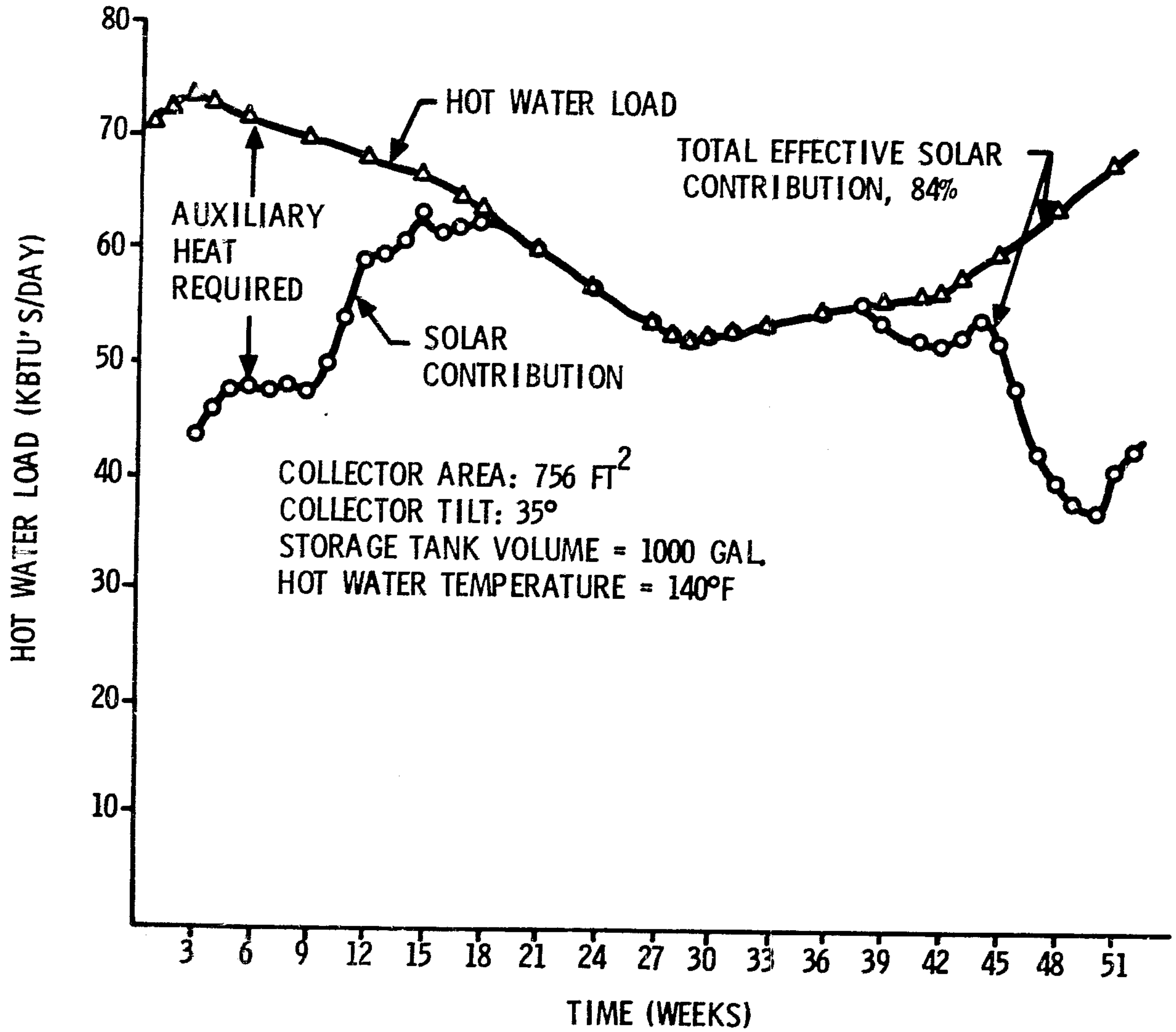

Figure 4-71. Weekly Solar Contribution to Hot Water Load - SFR

for the typical year, 1952, is 96.43 x 10^6 Btu, of which 56.95 x 10^6 Btu, or 59 percent, is supplied by solar energy. The yearly service hot water load is 22.5 x 10^6 Btu of which 18.89 x 10^6 or 84 percent is supplied by solar.

The annual cooling load is 36.15 x 10^6 Btu and the solar system in combination with the Rankine cycle driving a vapor compression air conditioner supply 16.8 x 10^6 Btu, or 46.5 percent. Figures 4-72 and 4-73 present the annual cooling and heating loads for the Atlanta SFR. Note that the Rankine cycle air conditioner operates most of the time (800 hours) at a house-cooling load of 4000 to 8000 Btu/hr, or about 0.5 ton. Total annual time that the air conditioner is on is 937 hours. Less than 5 hours occur at a house cooling load in excess of 3 tons (36,000 Btu/hr). An unusually high cooling load occurred in June, 1952. The heating loads shown in Figure 4-72 indicate less than 14 hours at a load in excess of 52,000 Btu/hr.

The operation of the pumps and fans for the solar HVAC system consumes 3805 kW-hr annually. This represents a cost of $135.46 per year based on an electrical charge of $0.356/kW-hr.

The following table presents the fuel costs in support of the solar HVAC system.

Heating System (required, 39 x 10^6 Btu)	
Electrical ($0.0356/kW-hr)	$412.13
Oil ($0.40/gallon)	140.99
Gas ($0.0017/$ft^3$)	83.89
Cooling System (required, 19.35 x 10^6 Btu)	
Electrical ($0.0356/kW-hr)	47.53
Hot Water System (required, 3.5 x 10^6 Btu)	
Electrical ($0.0356/kW-hr)	37.75
Oil ($0.40/gallon)	12.91
Gas ($0.0017 ft^3)	7.68

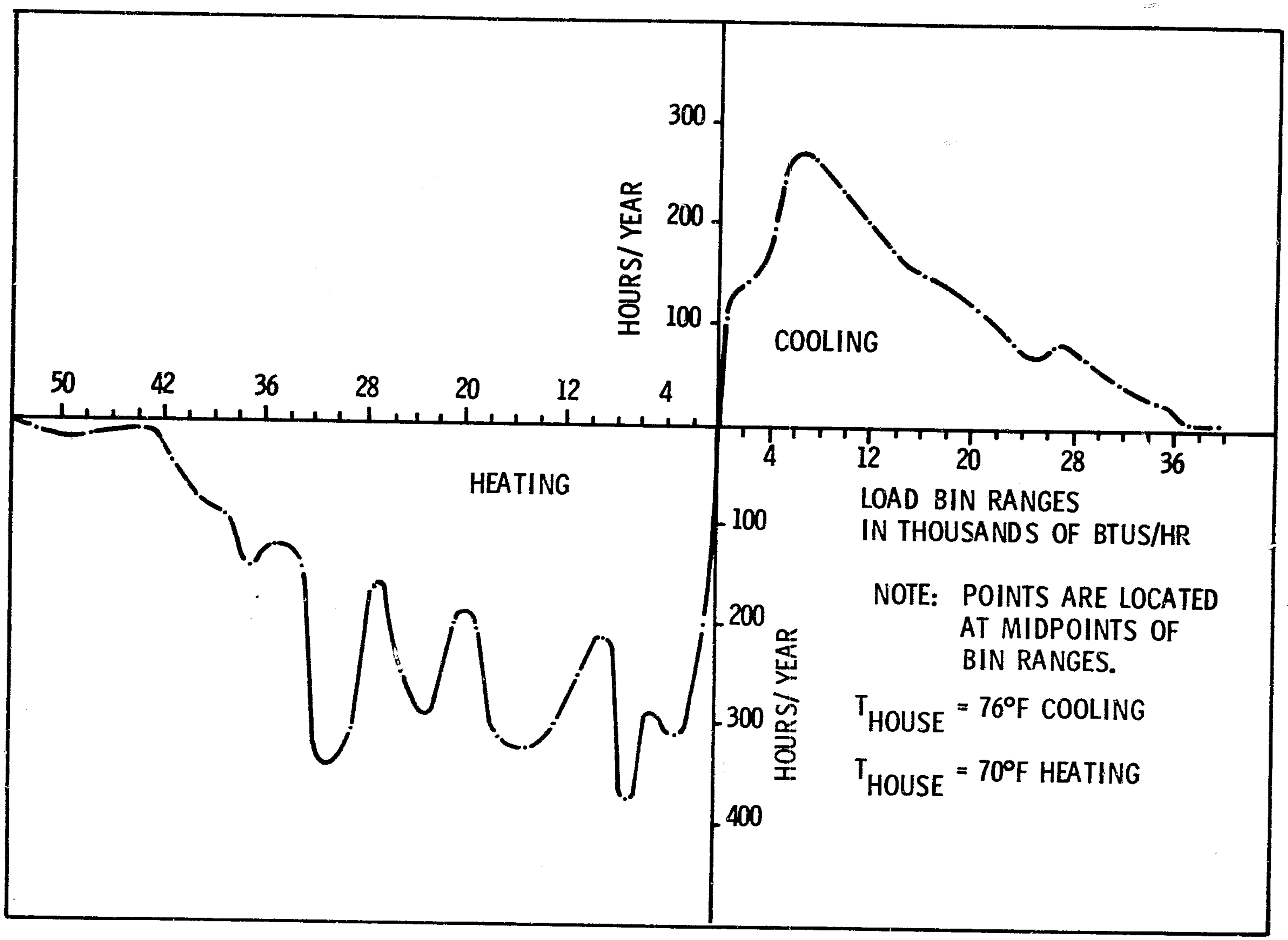

Figure 4-72. Cooling and Heating Load Histogram - SFR

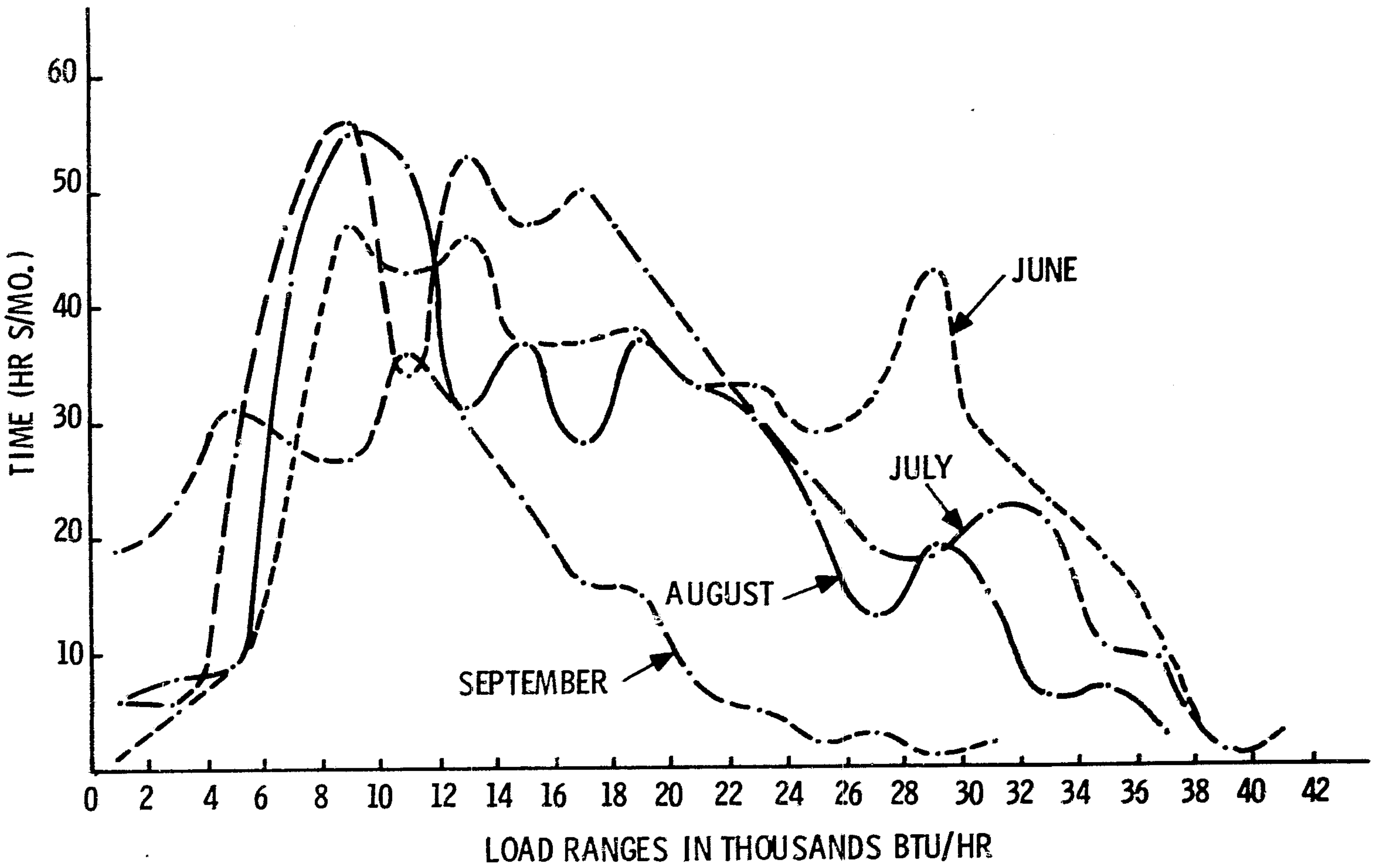

Figure 4-73. Cooling Load for SFR

The aforementioned fuel costs, plus the pumping and fan costs, represent the annual cost to the homeowner to heat, cool, and provide hot water to the home. An economic analysis has been made to determine the relative economic merits of the solar HVAC system in conserving conventional fuels and to determine the potential payback period for the capital investment associated with the solar system.

The estimated cost for just the baseline solar HVAC system is determined from the following equation.

Cost (including installation) = \$11,661 + \$17.85 (collector area)

For the baseline collector area of 756 ft^2, the total system cost for the Atlanta SFR is \$25,155.60. This represents an annual cost of \$2,374.44 based on a 20-year loan at 7 percent interest.

Assuming that all the auxiliary heating, cooling and hot water systems are based on electric power*, the total annual electric power requirement (including pumps and fans) in support of the solar HVAC system is \$632.87. If one adds this to the mortgage P&I and assumes that electrical power rates escalate at 10 percent/year over the next 20 years, the solid curve shown in Figure 4-74 can be constructed. This curve represents the future annual cost for the homeowner for the payoff of the solar HVAC system capital investment as well as the auxiliary electrical power costs to heat, cool and provide hot water.

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* This assumption is not considered unrealistic due to the following factors:

a) uncertainties in future gas reserves as well as gas rates make any analysis of this fuel source questionable

b) oil rates are subject to similar uncertainties as well as international politics

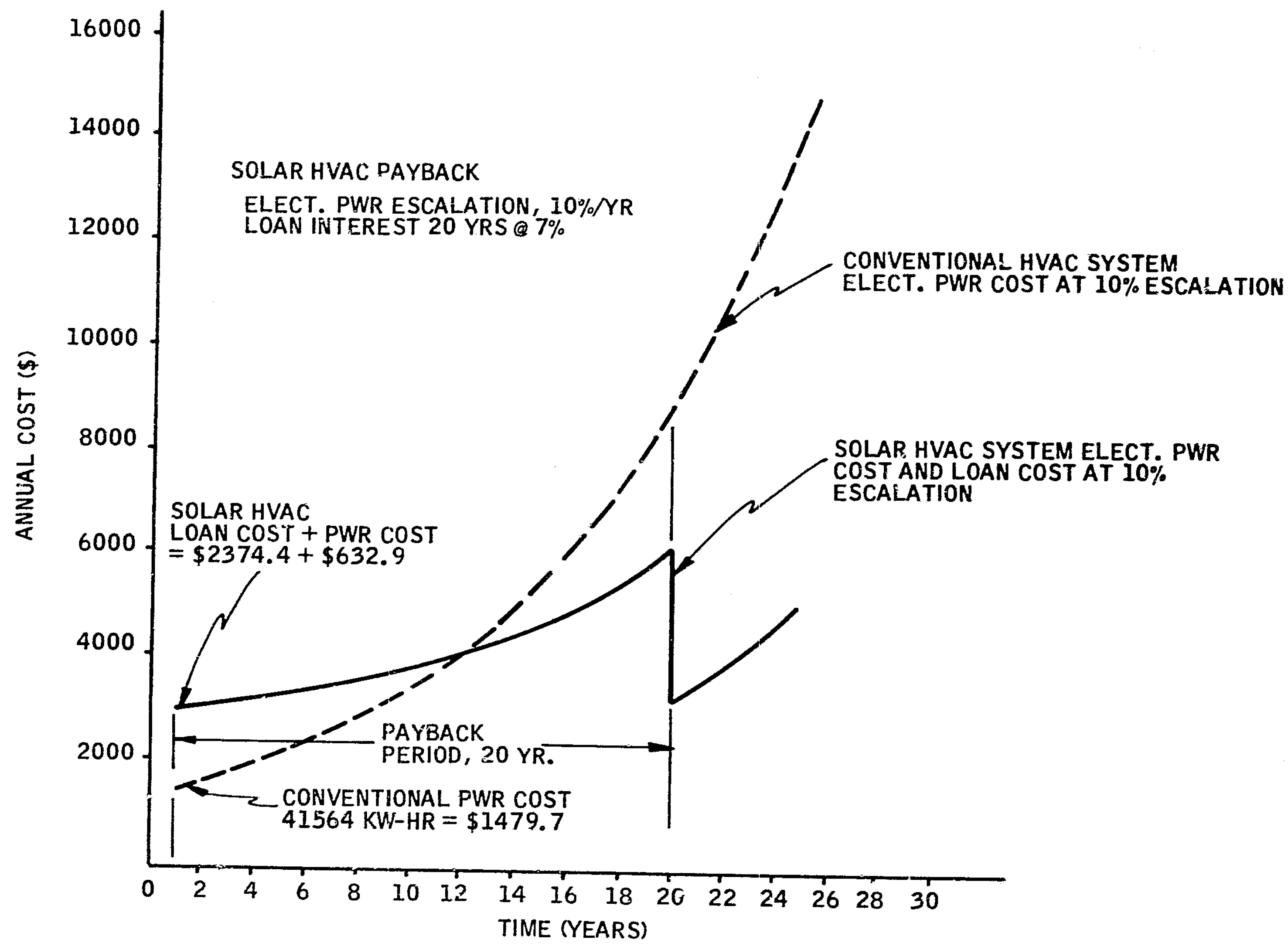

Figure 4-74. Solar HVAC Payback - SFR

The annual cost to meet the same total house loads with a conventional all-electric HVAC system is estimated below:

$$\text{Heating} = \frac{96.43 \times 10^6 \text{ Btu}}{3413 \text{ Btu/kW-hr}} = 28,253 \text{ kW-hr}$$

$$\text{Cooling} = \frac{36.15 \times 10^6 \text{ Btu}}{\text{EER* } = 6.1} = 5.926 \text{ kW-hr}$$

$$\text{Hot Water} = \frac{22.51 \times 10^6 \text{ Btu}}{3413 \text{ Btu/kW-hr}} = 6,595 \text{ kW-hr}$$

*EER (energy efficiency ratio) is estimated in ASHRAE 90-75 to be 6.1.

Total annual power requirement for the conventional HVAC system is 41,564 kW-hr or $1,479.68. Now if one escalates this annual cost by 10 percent, the dashed curve in Figure 4-74 is obtained. The area under these curves over 20 years represents a total dollar outlay. It is obvious that solar system payback is just over 20 years based on a comparison of areas.

Now one could discount these results by not accepting the 10 percent escalation figure. However, let us assume that this analysis is merely an example to demonstrate the sensitivities of solar HVAC system economic analysis. The curves indicate two very basic facts:

a) Solar HVAC system economic payback within a 20-year period will require significant fuel escalation rates in the future.

b) Reducing the initial system cost will also strongly reduce payback period.

As an example of the latter, consider that the cost of the baseline solar HVAC system for the SFR can be reduced as a result of mass production and cost reduction techniques.

Low-cost solar HVAC system for SFR --

R/C - A/C	\$3,000 each (assumed \$8,000 above)
Collector	\$6.00/ft^2 (assumed \$13.50/ft^2 above)
Collector Installation	\$4.35/ft^2
Remaining System	\$3.661.00 (heat exchangers, storage, etc.)

The "low-cost" solar HVAC system would be \$14,485.60 which is \$1,367.30 per year on a 20-year, 7 percent loan. Assuming that this system obtains the same thermal performance merits as the baseline system, the yearly cost figures for this system as compared to a conventional HVAC system are shown in Figure 4-75. At an electric power escalation factor of 10 percent the payback period for the "low-cost" solar HVAC is about 11 years. Of equal significance is a total power cost savings of over \$19,000.00 in the 20-year period. At an electric power escalation rate of 5 percent, the payback period increases to about 19 years.

In conclusion, solar HVAC systems for single-family residences can be shown to be economically competitive with reasonable payback time periods only if the following trends occur in the future:

- Fuel costs must <u>escalate</u> from 5 to 10 percent per year.
- A solar HVAC system that exhibits acceptable thermal performance must be designed, built and installed at <u>low cost</u>.

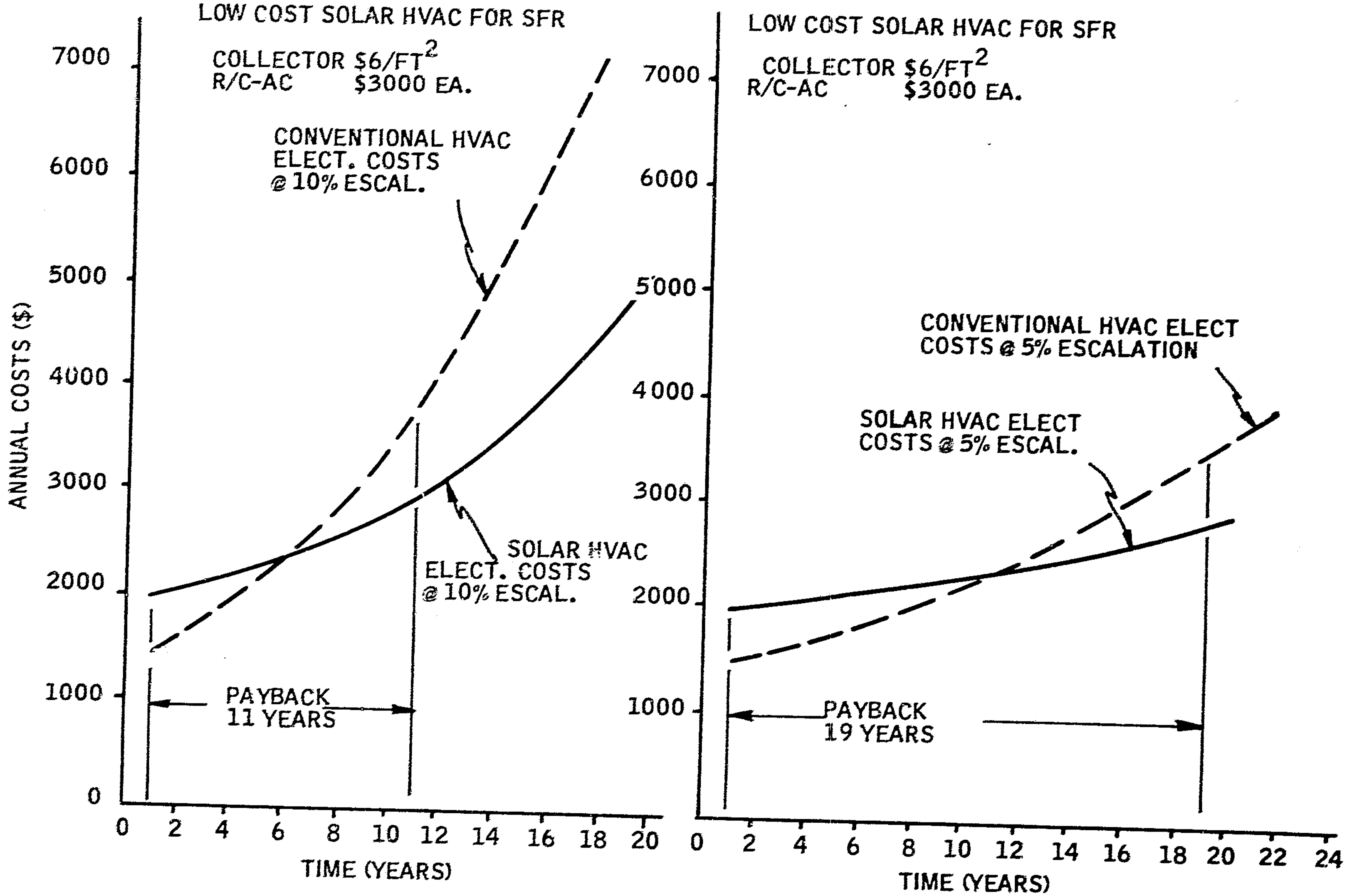

Figure 4-75. Low Cost Solar HVAC Payback - SFR

4.14.6.2 Multifamily Residence (MFR) -- The recommended solar-assisted heating, cooling and hot water system for the multifamily residence is a hydronic-to-warm air heating system, a Rankine cycle driving a 25-ton water chiller which operates at constant speed, and a domestic hot water preheat system. The baseline solar HVAC system consists of the major components shown in Figure 4-53 and listed in Table 4-7.

Table 4-7. Multifamily Residence (9,600 ft^2) Atlanta, Georgia

Baseline Solar HVAC System Design:

a) Cooling Subsystem

25-ton air conditioner
COP = 6
Constant speed control
Rankine cycle design point
20 hp at 190°F
η_{RC} at 190°F = 8.6 percent
Control range, 150°F - 200°F

b) Heating Subsystem

Capacity, 600,000 Btu/hr
Hx coil effectiveness, 0.6

c) Storage Subsystem

Tank capacity, 8,333 gallons
Storage Hx effectiveness, 0.55

d) Solar Collector Subsystem

Area 6300 ft^2 (total) 5227 ft^2 (effective)
Number of collectors, 350
Flow rate, 100 gpm

e) House Loads

UA = 4,416 Btu/hr°F
Infiltration, 3072 cfm (day and night)
Internal load, 25,500 Btu/hr (day) 13,500 Btu/hr (night)
Sun load on 480 ft^2 windows
Design day heating load, 349,987 Btu/hr based on
T_{AM} = 23°F, T_{House} = 70°F
Design day cooling load, 276,558 Btu/hr (23 tons) based on
T_{AM} = 92°F, T_{House} = 76°F

The thermal performance of this system for an Atlanta, Georgia, MFR is shown in Figures 4-76 through 4-78. Solar contributions to the three loads are summarized below.

	Annual Btu	Solar Supplied Btu	Solar Contribution
Heating Load	624.5×10^6	407.0×10^6	65%
Cooling Load	325.0×10^6	139.9×10^6	43%
Hot Water Load	283.2×10^6	238.3×10^6	84%

The annual heating and cooling load distributions for the Atlanta MFR are shown in Figure 4-79. The 25-ton Rankine cycle water chiller operates most of the time at 130,000 to 150,000 Btu/hr, or about 12 tons. The total annual time that the air conditioner operates is 1009 hours, with only 3 hours at a load over 300,000 Btu/hr, the 25-ton capacity of the cooling system.

The annual power consumption of the pumps and fans for this solar HVAC system is 42,678 kW-hr. This represents an annual cost of \$1,519.34 based on \$0.0356/kW-hr. The auxiliary fuel costs in support of the solar HVAC system are listed below:

Heating System (217.5×10^6 Btu)	
Electrical (\$0.0356/kW-hr)	\$2,271.11
Oil (\$0.40/gallon)	776.93
Gas (\$0.0017/ft^3)	462.28
Cooling System (185.1×10^6 Btu)	
Electrical (\$0.0356/kW-hr)	452.00
Hot Water (44.9×10^6 Btu)	
Electrical (\$0.0356/kW-hr)	468.33
Oil (\$0.40/gallon)	160.62
Gas (\$0.0017/ft^3)	95.33

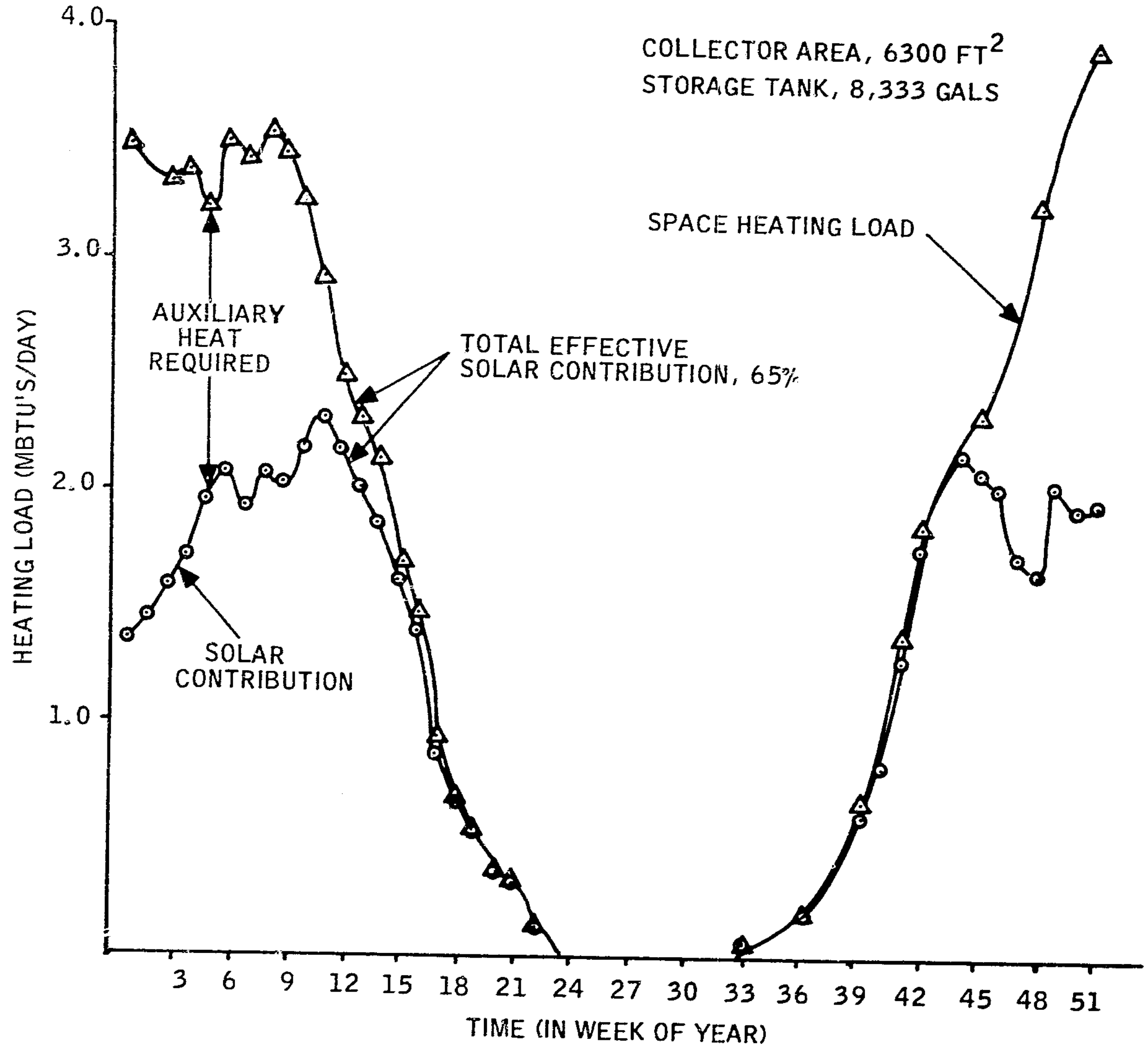

Figure 4-76. Weekly Solar Contribution to Heating Load - MFR

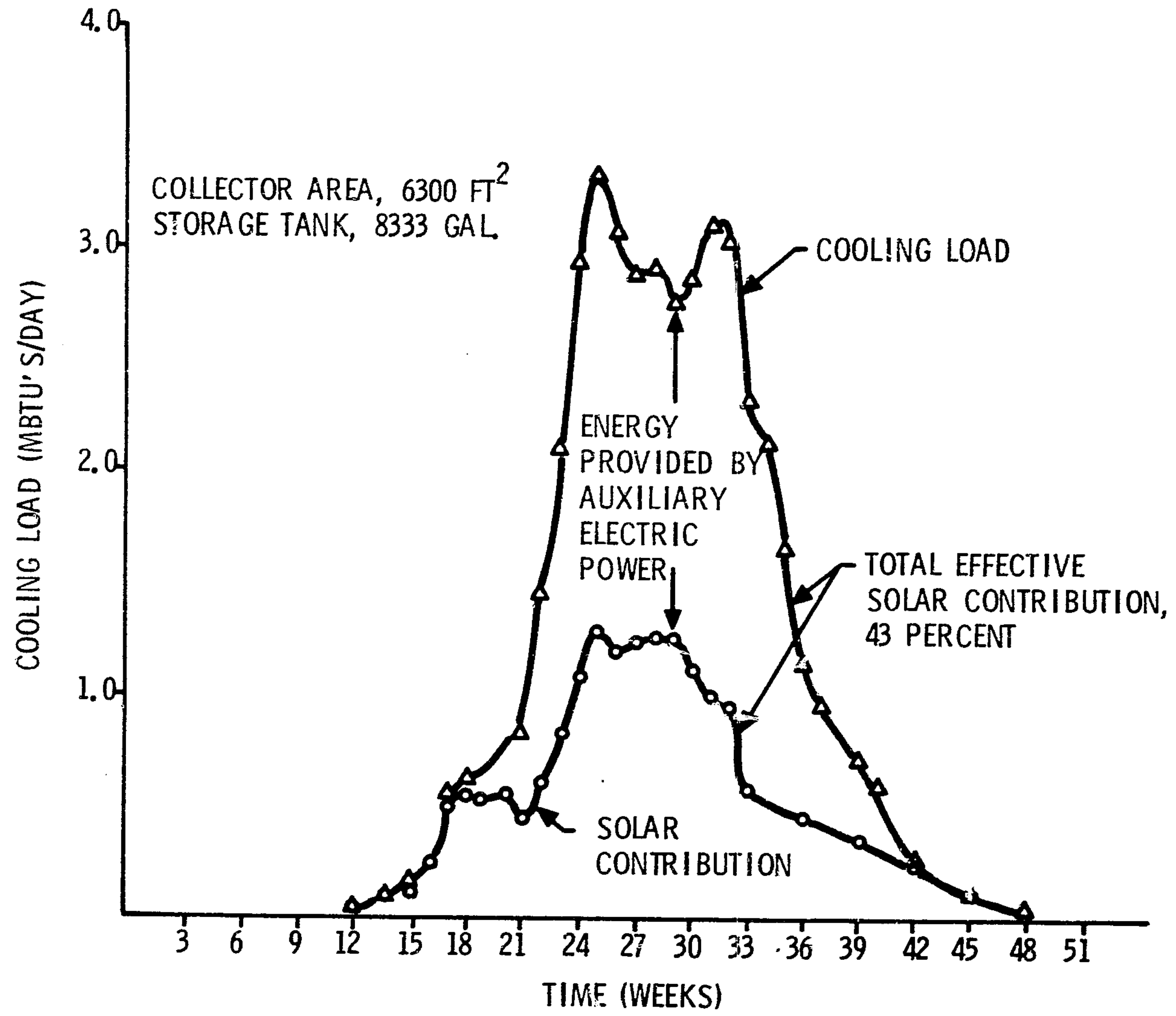

Figure 4-77. Weekly Solar Contribution to Cooling Load - MFR

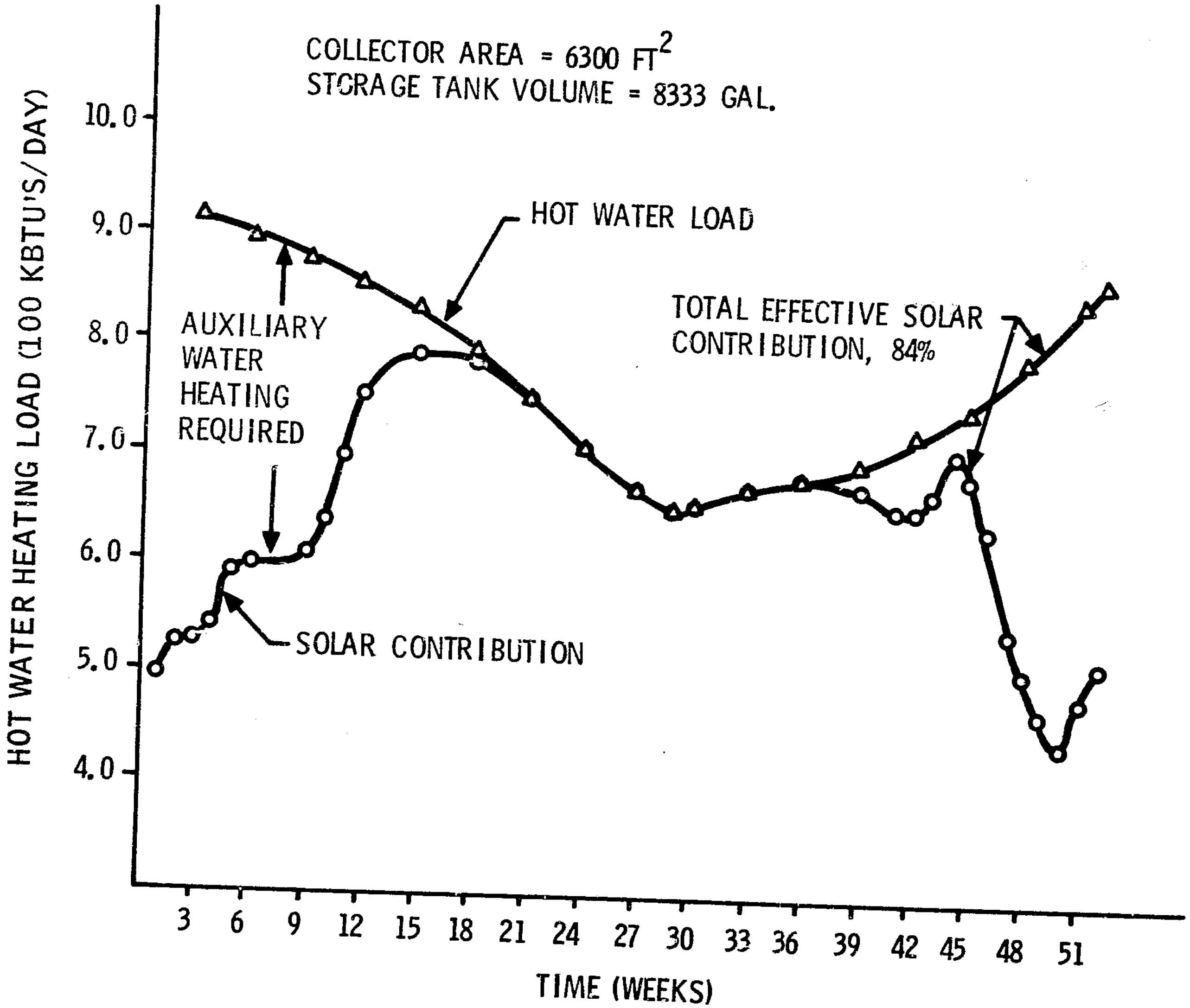

Figure 4-78. Weekly Solar Contribution to Hot Water Load - MFR

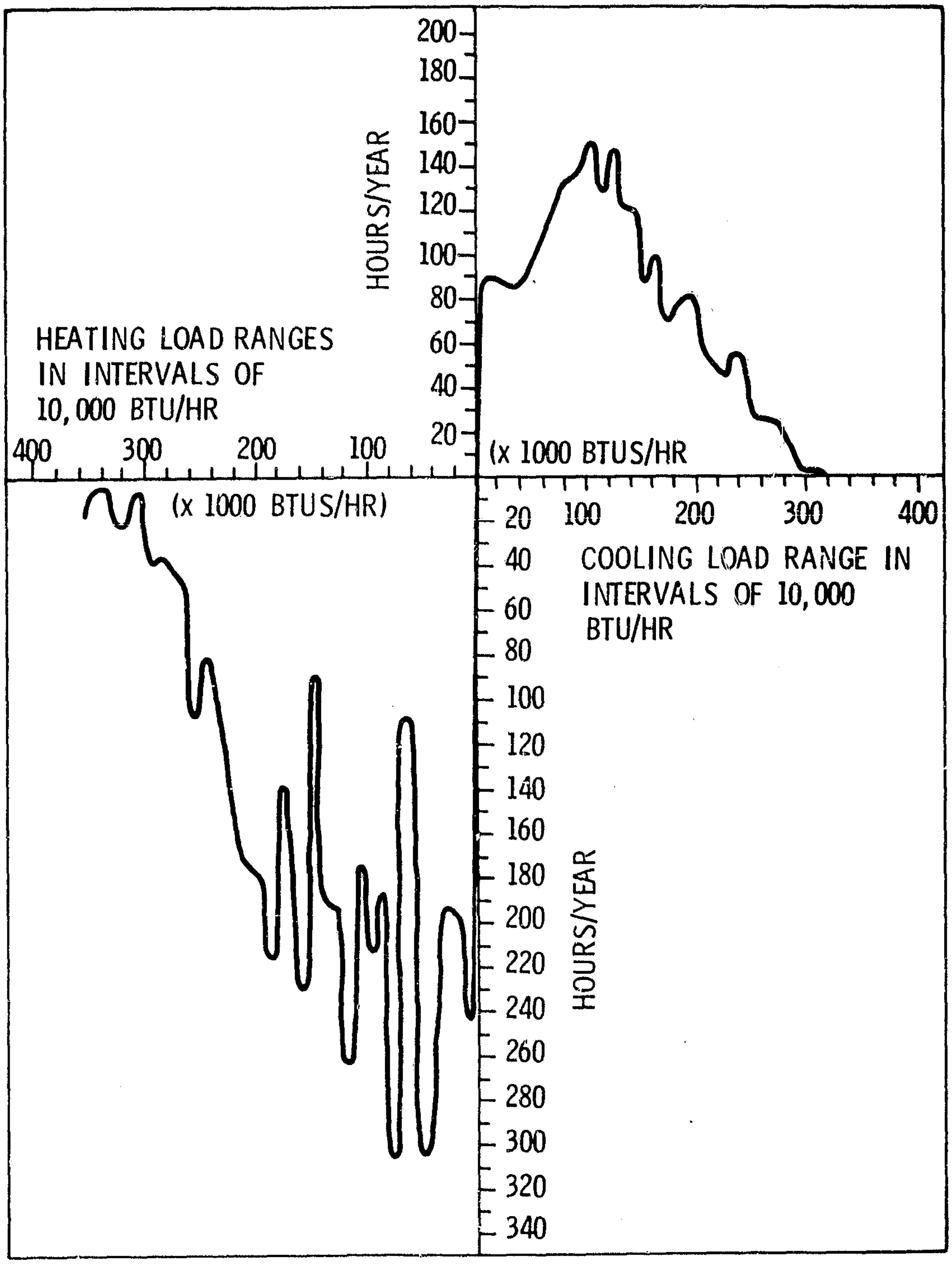

Figure 4-79. Cooling and Heating Load Histogram - MFR

An economic analysis of the MFR solar HVAC system has been performed in a method similar to the SFR. Total solar HVAC system cost is estimated from the following equation:

$$\text{Cost (including installation)} = 70,376 + 25.8\ (6,300\ \text{ft}^2)$$

$$= \$232,916.00$$

A 20-year loan at 7 percent interest represents an annual loan cost to the MFR owner of $21,984.94. An all-electric auxiliary heating, cooling and hot water system represents an annual cost of $4,710.78, including operating costs for the pumps and fans. The solid curve in Figure 4-80 represents the future annual costs of the solar HVAC system at an assumed 10 percent escalation rate in the cost of electricity.

A conventional all-electric HVAC system would require the following electric power.

$$\text{Heating} = \frac{624.5 \times 10^6\ \text{Btu}}{3413\ \text{Btu/kW-hr}} = 182,976\ \text{kW-hr}$$

$$\text{Hot Water} = \frac{283.2 \times 10^6\ \text{Btu}}{3413\ \text{Btu/kW-hr}} = 82,976\ \text{kW-hr}$$

$$\text{Cooling} = \frac{325 \times 10^6\ \text{Btu}}{6.8^{*}\ \text{Btu/kW-hr}} = 47,794\ \text{kW-hr}$$

$$\text{Total} \quad 313,747\ \text{kW-hr}$$

This represents an annual cost of $11,169.42 at an electric power rate of $0.0356/kW-hr. The dashed curve shown in Figure 4-80 represents the annual cost to operate the conventional HVAC system at the power escalation rate of 10 percent for the next 20 years. The system payback period appears to be 21 years. This time is in excess of the assumed 20-year operating life

*EER (energy efficiency ratio) is estimated in ASHRAE 90-75 to be 6.8.

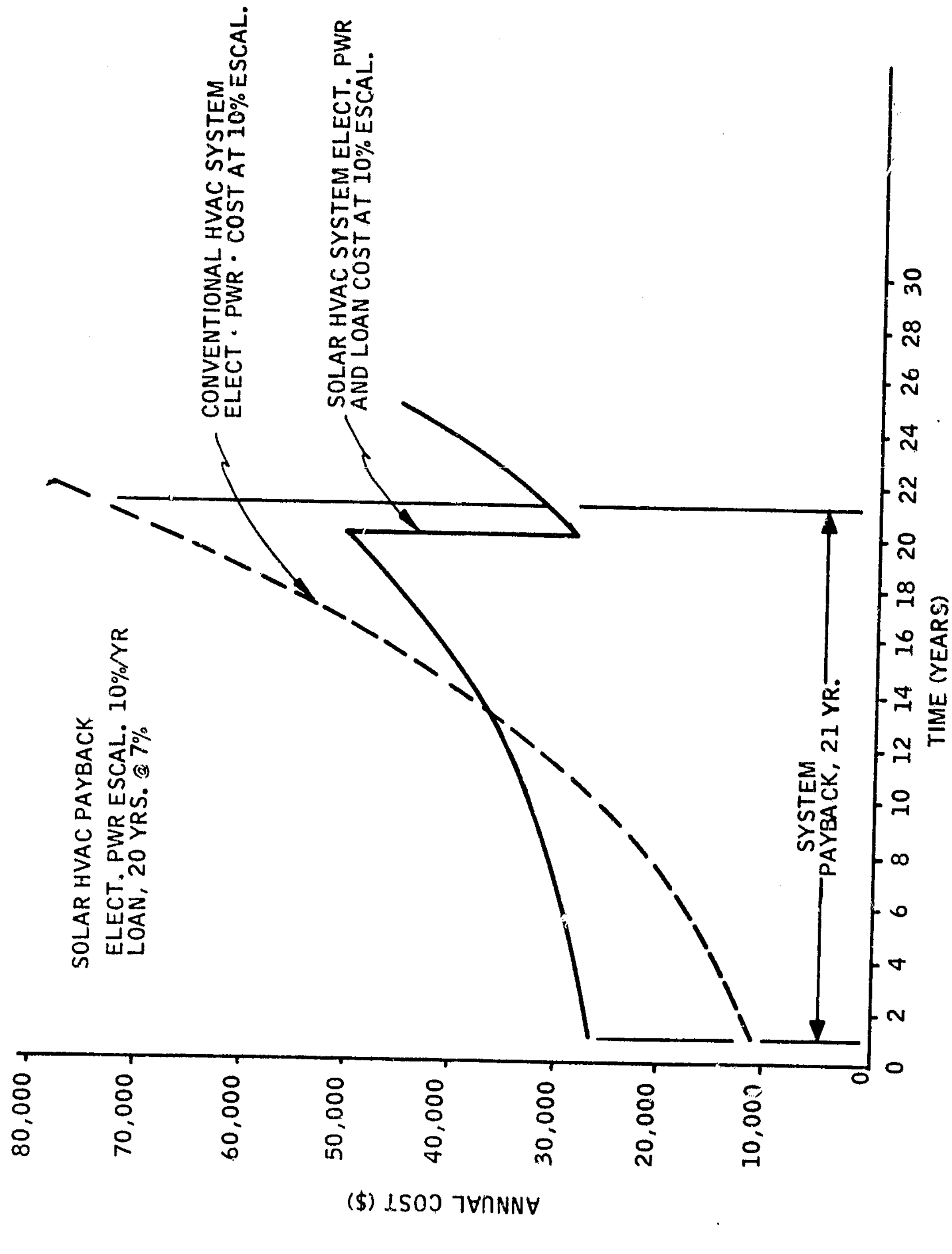

Figure 4-80. Solar HVAC Payback - MFR

for the solar HVAC system. However, this figure does illustrate that if solar HVAC systems demonstrate operational life times in excess of 20 years, a significant cost advantage will exist in the years following satisfaction of the loan.

No attempt has been made at this time to estimate a "low cost" solar HVAC system for the MFR; however, it is expected that conclusions similar to those reached for the SFR system would result:

> "Solar HVAC systems may be economically competitive with conventional systems provided: 1) their present fabrication and installation costs can be reduced; and 2) the cost of conventional power escalates from 5 percent to 10 percent per year."

4.14.6.3 Commercial Building -- The recommended solar-assisted heating, cooling and hot water system for the commercial building is an expansion of the MFR system. The major system parameters for the commercial building solar HVAC system are listed in Table 4-8. The cooling subsystem consists of three 25-ton Rankine cycle-driven water chillers instead of a large individual 75-ton R/C - A/C cooling subsystem.

The collector area of 12,060 ft^2 was selected to provide at least a 40 percent solar contribution to the cooling load. The thermal performance of the baseline system in meeting the building's heating, cooling and hot water loads is shown in Figures 4-81 through 4-83 and summarized as follows.

Table 4-8. Commercial Building (32,500 ft^2) Atlanta, Georgia

Baseline Solar HVAC System Design:

a) Cooling Subsystem

Three 25-ton air conditioners, total 75 tons
COP = 6
Constant speed control
Rankine design point
20 hp at 190°F
η_{RC} at 190°F = 8.6 percent
Control range, 150°F - 200°F

b) Heating Subsystem

Capacity, 800,000 Btu/hr
Hx coil effectiveness, 0.6

c) Storage Subsystem

Tank capacity, 15,600 gallons
Storage Hx effectiveness, 0.55

d) Solar Collector Subsystem

Area 12,060 ft^2 (total)   10,010 ft^2 (effective)
Number of collectors, 670
Flow rate, 300 gpm

e) Building Loads

UA = 12,340 Btu/hr°F
Infiltration, 225 cfm
Ventilation, 4,500 cfm (day)   0 cfm (night)
Internal load, 273,180 (day)   0 (night)
Design day heating load = 819,821 Btu/hr
Design day cooling load = 853,631 Btu/hr

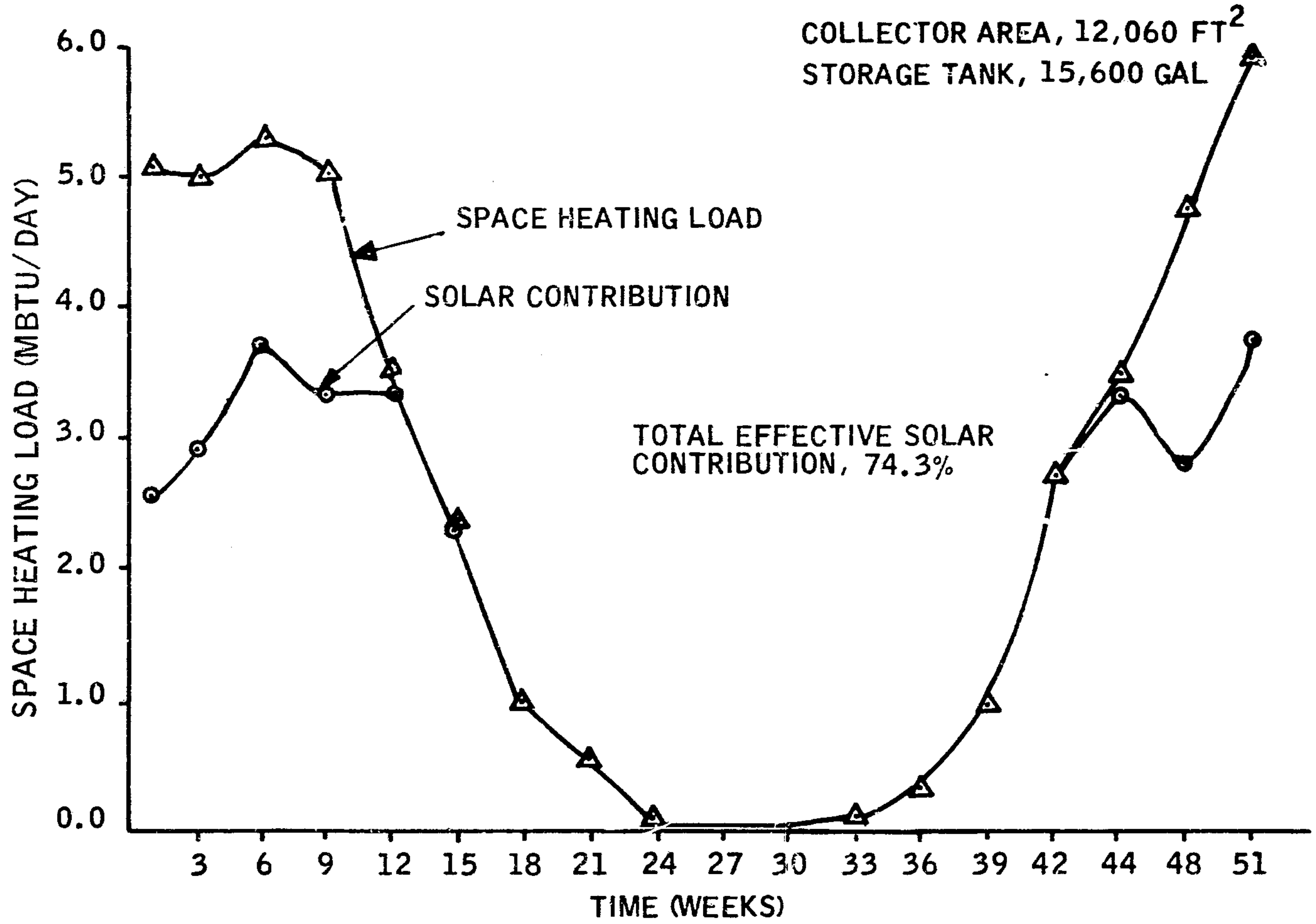

Figure 4-81. Weekly Solar Contribution to Heating Load - COM.

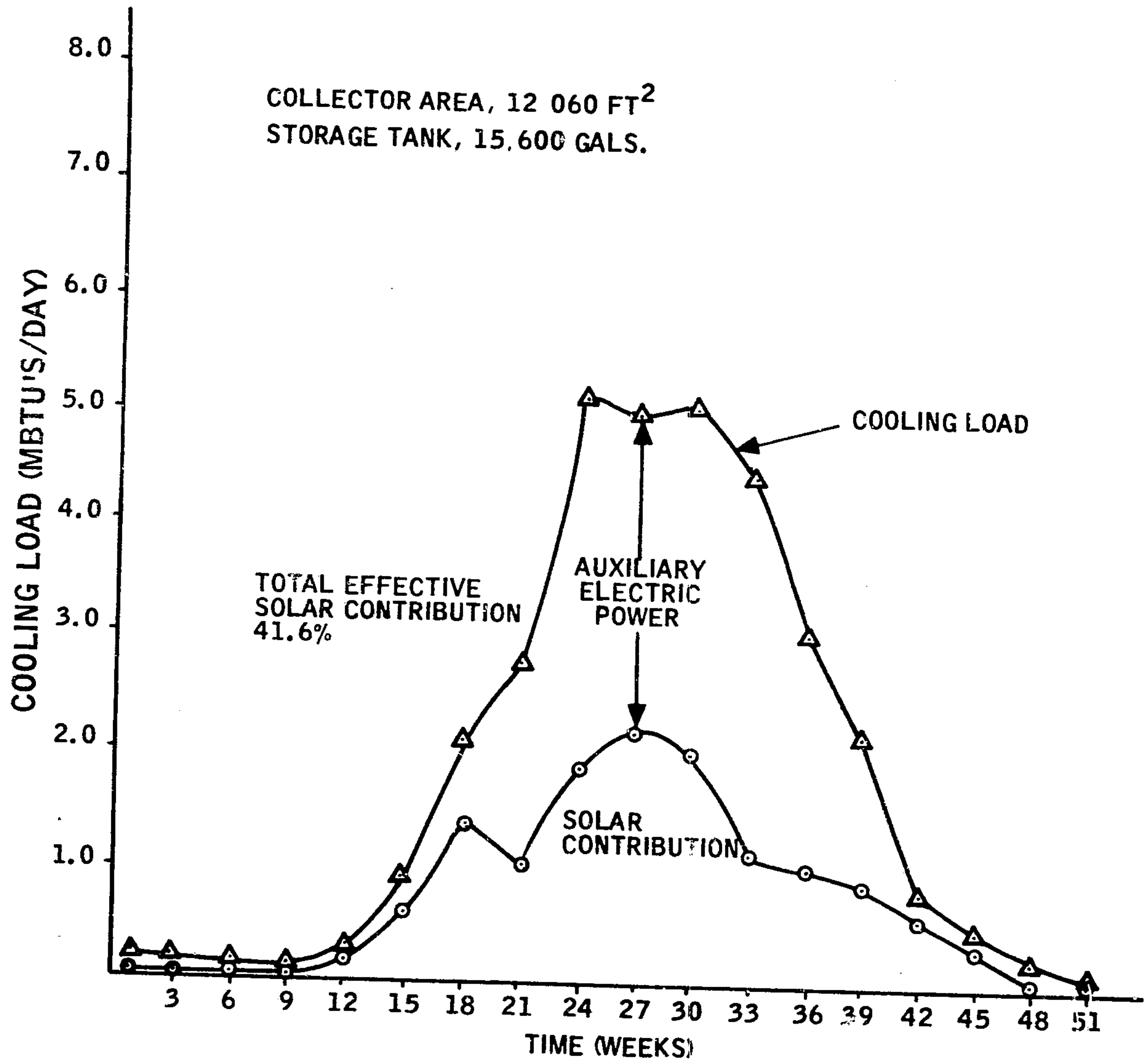

Figure 4-82. Weekly Solar Contribution to Cooling Load - COM.

2-3

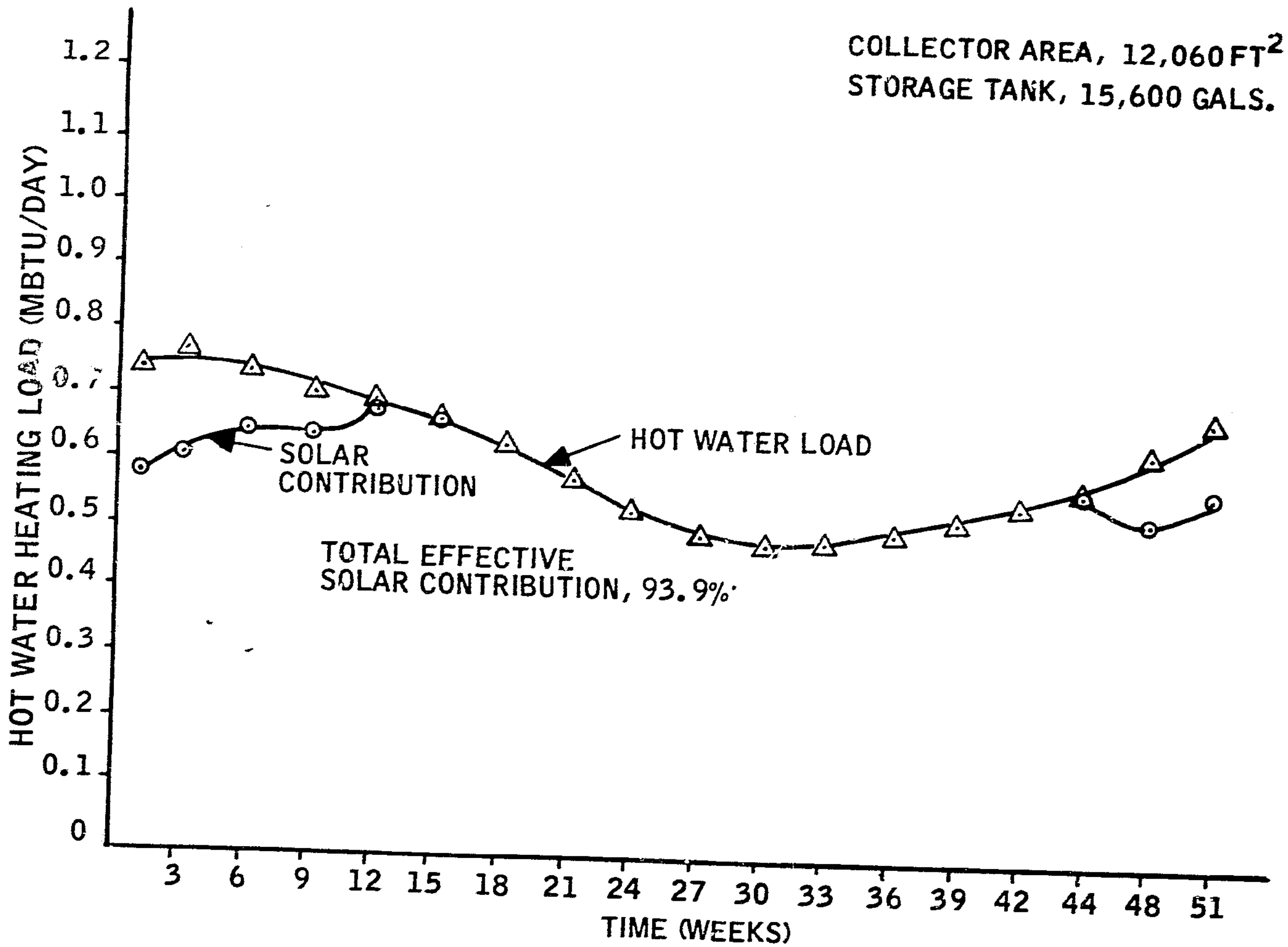

Figure 4-83. Weekly Solar Contribution to Hot Water Load - COM.

	Annual Load Btu	Solar Supplied Btu	Solar Contribution
Heating Load	925.3×10^6	687.5×10^6	74.3%
Cooling Load	702.7×10^6	292.4×10^6	41.6%
Hot Water Load	226.4×10^6	212.5×10^6	93.9%

T. e annual costs for conventional fuels to support the commercial building solar HVAC system are listed below.

Heating (237.8×10^6 Btu)	
Electrical ($0.0356/kW-hr)	$ 2482
Oil ($0.4/gallon)	844
Gas ($0.0017/ft^3)	505
Cooling (410.3×10^6 Btu)	
Electrical ($0.0356/kW-hr)	884
Hot Water (13.9×10^6 Btu)	
Electrical ($0.0356/kW-hr)	146
Oil ($0.4/gallon)	50
Gas ($0.0017/ft^3)	30

An analysis of the pumping and fan power required to operate this large solar HVAC system as well as an economic analysis has not been made as of this writing. The commercial building's internal loads and zone heating and cooling requirements are extremely site specific. Therefore, the economic analysis of this system will be performed at a time when more specific site information is available.

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